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PRACTICAL INSTRUCTIONS
FOR USING THE
STEAM ENGINE INDICATOR

CROSBY STEAM GAGE & VALVE CO.
BOSTON, MASS., U. S. A.

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PRACTICAL INSTRUCTIONS

RELATING TO THE CONSTRUCTION AND USE OF THE

STEAM ENGINE INDICATOR.

IN THREE PARTS.

PART FIRST.

GENERAL DESIGN AND CONSTRUCTION OF STEAM ENGINE
INDICATORS. SPECIAL DESIGN, CONSTRUCTION
AND USE OF THE

CROSBY INDICATOR,

WITH

DIRECTIONS FOR ITS ATTACHMENT AND THE CONSTRUCTION
OF SUITABLE MECHANISM FOR OPERATING THE DRUM,
TOGETHER WITH FULL INSTRUCTIONS FOR TAKING
DIAGRAMS, COMPUTING HORSE-POWERS, ETC.,

TO WHICH IS ADDED COMPLETE
DIRECTIONS FOR USING THE

POLAR PLANIMETER, AND PEABODY'S THROTTLING
CALORIMETER.

PART SECOND.

SOME ADDITIONAL USES OF THE INDICATOR.

PART THIRD.

PROPERTIES OF STEAM AND PERFECT GASES. CORRECT
METHODS FOR MAKING ENGINE TESTS WITH FULL
NUMERICAL CALCULATIONS.

Part Second and Chapters I. and II. of Part Three
BY EDWARD F. MILLER.
Chapter III. of Part Three
BY ALBERT F. HALL.

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INTRODUCTION.

The purpose of this book is to enable the engineer of ordinary ability to understand: First, The design, construction, and use of the Crosby Steam Engine Indicator. Secondly, To make suitable preparation for applying it to a steam engine, including the mechanism for operating the paper drum. Thirdly, To take diagrams, read them intelligently, and, after some experience, deduce from them such information as to the working of an engine, as a good instrument skilfully applied and handled is capable of revealing to the studious and observing mind.

CROSBY STEAM GAGE & VALVE CO.

Boston, Oct. 1, 1905.

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PRACTICAL INSTRUCTIONS

RELATING TO THE CONSTRUCTION
AND USE OF THE

STEAM ENGINE INDICATOR.

PART I.

CHAPTER I.

THE STEAM ENGINE INDICATOR.

The steam engine indicator, invented by James Watt, and long kept secret, was for many years after its secret became known, strangely neglected by most makers and users of steam engines.

The earlier forms of the instrument, which preceded that invented by Richards, were so imperfect and so ill-adapted to engines running at other than very low speeds, that their indications were often misleading, more often unintelligible, and seldom of much value beyond revealing the point of stroke at which the valves opened and closed: a most valuable service, alone worth the cost of an indicator, but only a small part of the service to be obtained from a really good instrument.

The general principles on which the best type of steam engine indicator is designed, may be briefly stated as follows: —

A piston of carefully determined area is nicely fitted into a cylinder so that it will move up and down without sensible friction. The cylinder is open at the bottom and fitted so that it may be attached to the cylinder of a steam engine and have free communication with its interior, by which arrangement the under side of the piston is subjected to all the varying pressures of the steam acting therein. The upward movement of the piston — due to the pressure of the steam — is resisted by a spiral spring within the cylinder, of known elastic force. A piston rod projects upward through the cylinder cap and moves a lever having at its free end a pencil point, whose vertical movement bears a constant ratio to that of the piston. A drum of cylindrical form and covered with paper is attached to the cylinder in such a manner that the pencil point may be brought in contact with its surface, and thus record any movement of either paper or pencil; the drum is given a horizontal motion coincident with and bearing a constant ratio to the movement of the piston of the engine. It is moved in one direction by means of a cord attached to the crosshead and in the opposite direction by a spring within itself.

When this mechanism is properly adjusted and free communication is opened with the cylinder of a steam engine in motion, it is evident that the pencil will be moved vertically by the varying pressure of steam under the piston, and as the drum is rotated by the reciprocating motion of the engine, if the pencil is held in contact with the moving paper during one revolution of the engine, a figure or diagram will be traced representing the pressure of steam in the cylinder; the upper line showing the pressure urging the piston forward, and the lower the pressure retarding its movement on the return stroke.

To enable the engineer to more correctly interpret the nature of the pressures, the line showing the atmospheric pressure is drawn, which indicates whether the pressure at any part is greater or less than that of the atmosphere.

From such a diagram may be deduced many particulars which are of supreme importance to engine builders, engineers, and the owners of steam plants.

WHAT IS THE GOOD OF AN INDICATOR?

This question was asked by a young engineer who had come to examine and purchase a Crosby indicator, with a view to rendering his services of greater value to his employer, by a knowledge and use of that instrument. His question was overheard by the proprietor of a large establishment in the city of Worcester, Mass., who took occasion to reply as follows: —

"I will tell you what good an indicator did at our works. Our steam engine was not giving sufficient power for our business, and we expected to be obliged to procure a larger one. A neighbor suggested that we have our engine indicated to see if we were getting the best service obtainable from it. This was done with a Crosby indicator, and the result was, that when the valves were properly adjusted and other slight changes made, we had *ample power*, and the improved condition of the engine made a reduction in our coal bills during the following year of $500."

Another case: An expert engineer was called to indicate several locomotives just completed by one of our prominent locomotive builders, who had in use a large Corliss engine, which had been running only a few months. When the locomotives were indicated, the proprietor proposed that the indicator be applied to the Corliss engine, the engineer of which remarked "Guess you'll find her all right, as she's running fine."

The first card showed that *nearly all the work was being done at one end of the cylinder.* The valves were changed and a great improvement was apparent in the running of the engine, while the actual consumption of coal was reduced from an average of 3370 pounds per day, before the change was made, to 2338 pounds afterwards.

These two instances are valuable in showing "the good of an indicator."

Items of Information to be obtained by the use of the Indicator.

The arrangement of the valves for admission, cut-off, release and compression of steam.

The adequacy of the ports and passages for admission and exhaust, and when applied to the steam chest, the adequacy of the steam pipes.

The suitableness of the valve motion in point of rapidity at the right time.

The quantity of power developed in the cylinder, and the quantity lost in various ways: by wire drawing, by back pressure, by premature release, by mal-adjustment of valves, leakage, etc.

It is useful to the designers of steam engines as by properly combining the cards, the effective pressure on the piston at any point can be determined, and from this the rotative effect or the rotating force acting at right angles to the crank, calculated.

Taken in combination with measurements of feed water and the condensation and measurement of the exhaust steam, with the amount of fuel used, the indicator furnishes many other items of importance when the economical generation and use of steam are considered.

For every one of these purposes it is important that the diagrams traced by the indicator should truly represent the path of the piston and the pressure exerted on both sides of the piston at every point of that path.

INDICATOR DIAGRAMS.

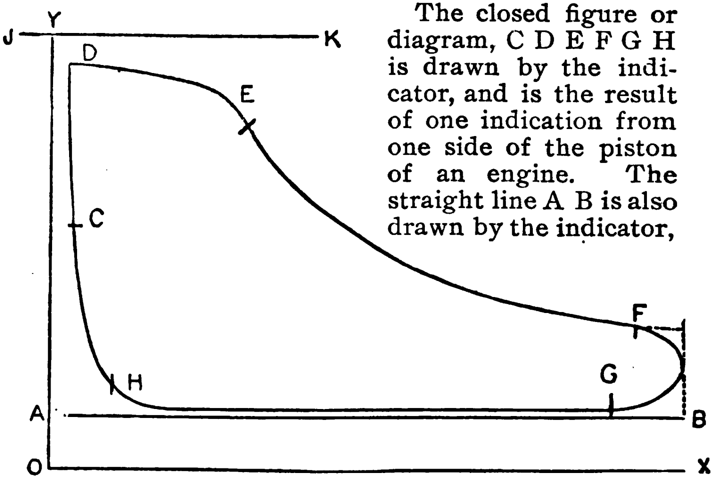
The degree of excellence to which steam engines of the present time have been brought is due more to the use of the indicator than to any other cause, as a careful study of indicator diagrams taken under different conditions of load, pressure, etc., is the only means of becoming familiar with the action of steam in an engine, and of gaining a definite knowledge of the various changes of pressure that take place in the cylinder.

An indicator diagram is the result of two movements, namely: a horizontal movement of the paper in exact correspondence with the movement of the piston, and a vertical movement of the pencil in exact ratio to the pressure exerted in the cylinder of the engine; consequently, it represents by its length the stroke of the engine on a reduced scale, and by its height at any point, the pressure on the piston, at a corresponding point in the stroke. The shape of the diagram depends altogether upon the manner in which the steam is admitted to and released from the cylinder of the engine; the variety of shapes given from different engines, and by the same engine under different circumstances, is almost endless, and it is in the intelligent and careful measurement of these that the true value of the indicator is found, and no one at the present day can claim to be a competent engineer who has not become familiar with the use of the indicator, and skilful in turning to practical advantage the varied information which it furnishes.

A diagram shows the pressure acting on one side of the piston only, during both the forward and return stroke, whereon all the changes of pressure may be properly located, studied, and measured. To show the corresponding pressures on the other side of the piston, another diagram must be taken from the other end of the cylinder. When the three-way cock is used, the diagrams from both ends are usually taken on the same paper, as in Fig. 2.

ANALYSIS OF THE DIAGRAM.

The names by which the various points and lines of an indicator diagram are known and designated are given on the page following. See Fig. 1.



The closed figure or diagram, C D E F G H is drawn by the indicator, and is the result of one indication from one side of the piston of an engine. The straight line A B is also drawn by the indicator,

FIG. 1.

but at a time when steam connection with the engine is closed, and both sides of the indicator piston are subjected to atmospheric pressure only.

The straight lines O X, O Y, and J K, when required, are drawn by hand as explained below, and may be called reference lines.

DIAGRAM LINES EXPLAINED.

(See Fig. 1.)

The admission line C D shows the rise or pressure due to the admission of steam to the cylinder by the opening of the steam valve. If the steam is admitted quickly when the engine is about on the dead-center this line will be nearly vertical.

The steam line D E is drawn when the steam valve is open and steam is being admitted to the cylinder.

The point of cut-off E is the point where the admission of steam is stopped by the closing of the valve. It is sometimes difficult to determine the exact point at which the cut-off takes place. It is usually located where the outline of the diagram changes its curvature from convex to concave.

The expansion curve E F shows the fall in pressure as the steam in the cylinder expands behind the moving piston of the engine.

The point of release F shows when the exhaust valve opens.

The exhaust line F G represents the loss of pressure which takes place when the exhaust valve opens at or near the end of the stroke.

The back pressure line G H shows the pressure against which the piston acts during its return stroke. On diagrams taken from non-condensing engines it is either coincident with or above the atmospheric line, as in Fig. 1. On cards taken from a condensing engine, however, it is found below the atmospheric line, and at a distance greater or less, according to the vacuum obtained in the cylinder.

The point of exhaust closure H is the point where the exhaust valve closes. It cannot be located very definitely, as the change in pressure is at first due to the gradual closing of the valve.

The compression curve H C shows the rise in pressure due to the compression of the steam remaining in the cylinder after the exhaust valve has closed.

The atmospheric line A B is a line drawn by the pencil of the indicator when its connections with the engine are closed, and both sides of the piston are open to the atmosphere. This line represents on the diagram the pressure of the atmosphere, or zero of the steam gage.

REFERENCE LINES EXPLAINED.

The zero line of pressure, or line of absolute vacuum O X, is a reference line, and is drawn by hand, 14$\frac{7}{10}$ pounds by the scale, below and parallel with the atmospheric line. It represents a perfect vacuum, or absence of all pressure

The line of boiler pressure J K is drawn by hand parallel to the atmospheric line and at a distance from it, by the scale equal to the boiler pressure shown by the steam gage. The difference in pounds between it and the line of the diagram D E shows the pressure which is lost after the steam has flown through the contracted passages of the steam pipes and the ports of the engine.

The clearance line O Y is another reference line drawn at right angles to the atmospheric line and at a distance from the end of the diagram equal to the same per cent. of its length as the clearance bears to the piston travel or displacement. The distance between the clearance line and the end of the diagram represents the *volume* of the clearance and waste room of the ports and passages at that end of the cylinder.

DERANGED VALVE MOTION.

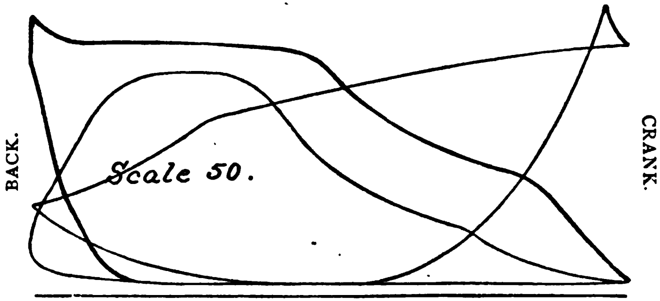


FIG. 2.

Fig. 2 shows two diagrams, one from each end of the cylinder of a single valve high pressure engine. This valve admits the steam over its ends and exhausts inside. The derangement is caused by the valve stem being too long; consequently, at the back end the diagram shows that the steam was admitted late, cut off early, exhausted early, and the exhaust valve closed late, so that there is little or no compression. The diagram at the crank end shows the opposite defects, viz.: — Steam is admitted too soon and carried too far on the stroke, the exhaust valve is opened too late and closed too soon to get the steam well out of the cylinder, causing excessive back pressure, — even greater than the boiler pressure as shown by the loop at the top.

To remedy this derangement, the valve stem should be shortened by the screw threads at one end. It may then be found that the steam valve opens a little too late at both ends, and it will therefore be necessary to turn the eccentric ahead on the shaft until both diagrams resemble the figures shown in the heaviest lines.

UNITS OF MEASUREMENT AND TECHNICAL TERMS.

All substances of whatever nature are measurable, and their measurements are referable to some established unit, to be properly expressed and dealt with. An intimate knowledge of some of these is indispensable to the engineer; a few are here briefly defined.

The Unit of Linear Measurement is the inch or one-twelfth part of a foot.

The Unit of Superficial Measurement is the square inch.

The Unit of Solid Measurement is the cubic inch.

The Unit of Fluid Pressure is the pound avoirdupois, consisting of 7,000 grains.

The Unit of Elasticity, or the pressure exerted by elastic fluids, is, for popular use, one pound on one square inch.

The Unit of Work or Power is one pound lifted twelve inches, or, in other words, one pound of force acting through one foot of distance, and is called the foot-pound.

Horse-Power. The standard used for measuring the power of a steam engine is the horse-power. It was originally determined by James Watt from experiments made on London dray-horses. It is considerably above the power of an ordinary horse, and is now simply an arbitrary standard. It is equal to 33,000 foot-pounds exerted during one minute of time, or 550 foot-pounds during one second. As a foot-pound is the amount of work done in raising one pound through the distance of one foot, an equivalent amount of work would be raising half a pound two feet, or twelve pounds one inch.

Indicated Horse-Power is the horse-power of an engine as found by the use of a steam engine indicator, and is thus expressed: I. H. P.

Net Horse-Power is the indicated horse-power of an engine, less the horse-power which is consumed in overcoming its own friction.

Wire drawing, as applied to steam, is the reducing of its pressure, due to its flowing through restricted or crooked pipes and passages.

Absolute Pressure is pressure reckoned from absolute vacuum; in other words, it is the pressure of any fluid as shown by a pressure gage, with the weight or pressure of the atmosphere added thereto.

Initial Forward Pressure in a cylinder is the pressure acting on the piston at or near the beginning of the forward stroke.

Terminal Forward Pressure is the pressure above the line of perfect vacuum that would exist at the end of the stroke if the steam had not been released earlier. It may be found by continuing the expansion curve to the end of the diagram, as in Fig. 1 at F, or it may be taken at the point of release. This pressure is always measured from the line of perfect vacuum, hence it is the absolute terminal pressure.

Mean Effective Pressure (written as M. E. P.) is that equivalent constant pressure which will do the same amount of work on the piston per stroke as is done by the varying pressure shown by the indicator card.

This may be calculated by dividing the area of the card by the length and multiplying by the scale of the spring used in the indicator, or it may be obtained by taking the average of a number of pressures measured between the top and the bottom lines of the card and at equidistant intervals along the length of the card. This is illustrated more fully later on.

Piston Displacement is the space in the cylinder swept through by the piston in its travel. It is reckoned in cubic feet, and is found by multiplying the net area of the piston in square feet by the length of stroke in feet, allowance being made for the piston rod.

Clearance is all the waste room or space at either end of the cylinder, between its head and the piston when on a dead center, including the counterbore and the ports, up to the face of the closed valves.

Clearance is generally given as a percentage of the piston displacement. Many of the modern steam engines have clearances no greater than 2¼ per cent.

The Number of Expansions is the number of times the steam admitted up to cut-off in the first cylinder of a multiple expansion engine has increased in volume up to release in the last cylinder.

If the release be considered to come at the end of the stroke the number of expansions for any engine may be calculated from the diameter of the cylinders and the per cent. of cut-off in the first cylinder.

Thus — cylinders of a triple engine are 9″ — 16″ — 24″ × 30″ stroke; cut-off at ⅓ stroke in high pressure cylinder.

As the strokes are the same on all the cylinders of this engine the volumes of the cylinders will be as the squares of their diameters.

If the high pressure cylinder took steam to the end of its stroke the number of expansions would be $\frac{24^2}{9^2}$, but the steam expands three times in the high, so the total expansion in the engine is $\frac{24^2}{9^2} \times 3 = 21.3$.

Sensible Heat is the temperature of any body, as air, water, or steam, which may be measured by the thermometer.

Specific Heat is the quantity of heat required to raise one unit of weight of the substance through one degree of temperature, measured in thermal units.

The Unit of Heat, or thermal unit, is the quantity of heat required to raise the temperature of one pound of water from 62° to 63° F.

Mechanical Equivalent of Heat. It has been found by experiment that if one pound of pure water at 62° F. be raised to 63° F. that energy is exerted equivalent to lifting seven hundred and seventy eight (778) pounds one foot high, or one pound seven hundred and seventy eight (778) feet high. This energy is called the *Mechanical equivalent of one thermal unit of heat*, and it is usually designated by the letter J and its reciprocal, or $\frac{1}{778}$ by A.

Saturated Steam. When steam is formed in a closed vessel in contact with its own liquid, it is said to be saturated and it will have a certain definite pressure and density corresponding to each different temperature. If, at the same time, the steam contains no liquid in suspension, it is said to be *dry* and *saturated*.

Superheated Steam. If after *all* the liquid has been converted into steam, more heat be added, the temperature will rise and the steam is said to be superheated, because its temperature will be greater than that corresponding to *saturated* steam of the *same pressure.* This difference is the number of degrees of superheat.

Priming. It is possible for saturated steam to hold particles of water in suspension. The steam is then said to be primed or wet. The amount of water so suspended may vary from zero to five per cent. Almost all boilers in which the steam is not in contact with the hot gases give steam primed from .1 per cent to 3 per cent.

Thermal Efficiency of an Engine. An engine having a thermal efficiency of 100 per cent. would require $\frac{33000}{778} = 42.42$ B. T. U. per H. P. per minute.

An engine which used 250 B. T. U. per H. P. per minute thus has a thermal efficiency of $\frac{42.42}{250} = .169$ or 16.9 per cent.

Mechanical Efficiency of an Engine. This is the ratio of the power delivered at the fly-wheel or shaft to that calculated from the indicator cards. This ratio is about 90 per cent.

CHAPTER II.

THE CROSBY STEAM ENGINE INDICATOR.

The Crosby Steam Engine Indicator is designed and constructed to meet the exacting requirements of modern steam engineering. During the last few years, under the keen search and exhaustive tests of eminent engineers, the practice in this department of science has undergone important changes, tending to establish more correct methods and thereby to reach more accurate results; especially is this true in the use and scope of the indicator, so that the work done with this instrument in former times seems coarse and crude when compared with the more exact attainment of the present.

Educators in the scientific schools of both Europe and America have seen the importance of more exact knowledge and instruction in the technical sciences; and the great achievements of recent years in the construction of buildings, ships, armaments, and machines attest the thoroughness with which research in these departments has been prosecuted; in none has there been greater progress made than in those of mechanical and steam engineering.

A knowledge of these facts has kept us on the alert in the manufacture of all our steam appliances, and especially in that of the steam engine indicator. Within a recent time we have, without any great change in its outward appearance, made important improvements, which, as we believe, place it far in advance of any other instrument of its kind. Slight changes in design, a more perfect mechanical construction due to the use of improved and specialized machinery, and a careful selection of metals for the different parts, have all contributed to this favorable result.

The movements of piston and pencil point are perfectly parallel, the movement of the pencil point is also exactly parallel with the axis of the drum.

The rating of the springs by our newly constructed testing apparatus, which embodies all the valuable aids to exactness which have yet been discovered, is nearer perfection than could have been attained, or even expected, until within a very recent time.

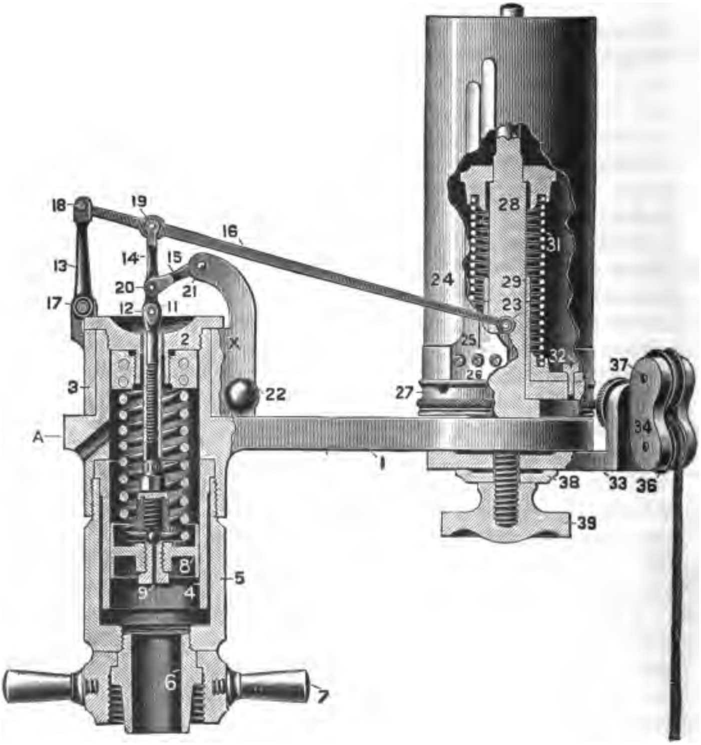
Description of the Indicator.

The illustration shows the design and arrangement of the parts of the Crosby Steam Engine Indicator.

Part 4 is the cylinder proper, in which the movement of the piston takes place. It is made of a special alloy, exactly suited to the varying temperatures to which it is subjected, and secures to the piston the same freedom of movement with high pressure steam as with low; and as its botton end is free and out of contact with all other parts, its longitudinal expansion or contraction is unimpeded, and no distortion can possibly take place.

Between the parts 4 and 5 is an annular chamber, which serves as a steam jacket; and, being open at the bottom, can hold no water, but will always be filled with steam of nearly the same temperature as that in the cylinder.

The Piston, 8, is formed from a solid piece of the finest tool steel. Its shell is made as thin as possible consistent with proper strength. It is hardened to prevent any reduction of its area by wearing, then ground and lapped to fit (to the ten-thousandth part of an inch) a cylindrical gage of standard size. Shallow channels in its outer surface provide a steam packing, and the moisture and oil which they retain act as lubricants, and prevent undue leakage by the piston. The transverse web near its center supports a central socket, which projects both upward and downward; the upper part is threaded inside to receive the lower end of the piston-rod; the upper edge of this socket is formed to fit nicely into a circular channel in the under side of the shoulder of the piston-rod, when they are properly connected. It has a longitudinal slot which permits the straight portion of wire at the bottom of the spring, with its bead, to drop to a concave bearing in the upper end of the piston-screw 9, which is closely threaded into the lower part of the socket; the head of this screw is hexagonal, and may be turned with the hollow wrench which accompanies the Indicator.



The Piston-rod, 10, is of steel, and is made hollow for lightness. Its lower end is threaded to screw into the upper socket of the piston. Above the threaded portion is a shoulder having in its under side a circular channel formed to receive the upper edge of the socket, when these parts are connected together. When making this connection BE SURE that the piston-rod is screwed into the socket as far as it will go, that is, until the upper edge of the socket is brought firmly against the bottom of the channel in the piston-rod. This is very important, as it insures a correct alignment of the parts, and a free movement of the piston within the cylinder.

The Swivel Head, 11, is threaded on its lower half to screw into the piston-rod more or less, according to the required height of the atmospheric line on the diagram. Its head is pivoted to the piston-rod link of the pencil mechanism.

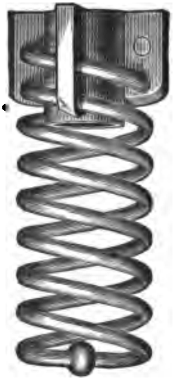
The Cap, 2, rests on top of the cylinder, and holds the sleeve and all connected parts in place. It has a central depression in its upper surface, also a central hole, furnished with a hardened steel bushing, which serves as a very durable and sure guide to the piston-rod. It projects downward into the cylinder in two steps, having different lengths and diameters; both these and the hole have a common center. The lower and smaller projection is screw-threaded outside to engage with the like threads in the head of the spring, and hold it firmly in place. The upper and larger projection is screw-threaded on its lower half to engage with the light threads inside the cylinder; the upper half of this larger projection, — being the smooth, vertical portion, — is accurately fitted into a corresponding recess in the top of the cylinder, and forms thereby a guide by which all the moving parts are adjusted and kept in correct alignment, which is very important, but which is impossible to secure by the use of screw threads alone.

The Sleeve, 3, surrounds the upper part of the cylinder in a recess formed for that purpose, and supports the pencil mechanism; the arm, X, is an integral part of it. It turns around freely, and is held in place by the cap. The handle for adjusting the pencil point is threaded through the arm, and being in contact with a stop-screw in the plate, I, may be delicately adjusted to the surface of the paper on the drum. It is made of hard wood in two sections; the inner one may be used as a lock-nut to maintain the adjustment.

The Pencil Mechanism is designed to afford sufficient strength and steadiness of movement, with the utmost lightness; thereby eliminating as far as possible the effect of momentum, which is especially troublesome in high speed work. Its fundamental kinematic principle is that of the pantograph. The fulcrum of the mechanism as a whole, the point attached to the piston-rod, and the pencil point are always in a straight line. This gives to the pencil point a movement exactly parallel with that of the piston. The movement of the spring throughout its range bears a constant ratio to the force applied, and the amount of this movement is multiplied six times at the pencil point. The pencil lever, links, and pins are all made of hardened steel; the latter, — slightly tapering, — are ground and lapped to fit accurately, without perceptible friction or lost motion.

Springs. In order to obtain a correct diagram, the height of the pencil of the indicator must exactly represent in pounds per square inch the pressure on the piston of the steam engine at every point of the stroke; and the velocity of the surface of the drum must bear at every instant a constant ratio to the velocity of the piston. These two essential conditions have been attained to a greater degree of exactness in the Crosby Indicator than in any other make, by a very ingenious construction and nice adaptation of both its piston and drum springs.

The Piston Spring is of unique and ingenious design, being made of a single piece of the finest spring steel wire, wound from the middle into a double coil, the ends of which are screwed into a brass head having four radial wings with helically drilled holes to receive and hold them securely in place.



Adjustment is made by screwing them into the head more or less, until exactly the right strength of spring is obtained, when they are there firmly fixed. This method of fastening and adjusting removes all danger of loosening coils, and obviates all necessity for grinding the wires, — a practice fatal to accuracy in indicator springs.

The foot of the spring, — in which lightness is of great importance, it being the part subject to the greatest movement, — is a small steel bead, firmly "staked" on to the wire. This takes the place of the heavy brass foot used in other indicators, and reduces the inertia and momentum at this point to a minimum, whereby a great improvement is effected. This bead has its bearing in the center of the piston, and in connection with the lower end of the piston-rod and the upper end of the piston-screw, 9 (both of which are concaved to fit), it forms a ball and socket joint which allows the spring to yield to pressure from any direction without causing the piston to bind in the cylinder, which is sure to occur when the spring and piston are rigidly united, as is the case in other indicators. Designing the spring so that any lateral movement it may receive when being compressed shall not be communicated to the piston and cause errors in the diagram, is of extreme importance.

The Drum Spring, 31, in the Crosby Indicator is a short helix, while in every other make a long volute spring is used.

It is obvious from the large contact surfaces of a long volute spring that its friction would be greater than that of a short, open helical form; also, that in a spring of its kind, for a given amount of compression,—as in the movement of an indicator drum,—the recoil would be greater and expended more quickly in the helical than in the spiral form.

If the conditions under which the drum spring operates be considered, it will readily be seen, that at the beginning of the stroke, when the cord has all the resistance of the drum and spring to overcome, the spring should offer less resistance than at any other time; in the beginning of the stroke in the opposite direction, however, when the spring has to overcome the inertia and friction of the drum, its energy or recoil should be greatest.

These conditions are fully met in the Crosby Indicator; its drum spring being a short helix having no friction, a quick recoil, and being scientifically proportioned to the work it has to do. At the beginning of the forward stroke it offers to the cord only a very slight resistance, which gradually increases by compression, until at the end its maximum is reached. At the beginning of the stroke, in the other direction, its strength and recoil is greatest, at the moment when it is most needed, and gradually decreases as the work it has to do decreases, until at the end of the stroke it is reduced to its minimum again. Thus, by a most ingenious balancing of opposing forces, a nearly uniform stress on the cord is maintained throughout each revolution of the engine.

The Drum, 24, and its appurtenances, except the drum spring, are similar in design and function to like parts of other indicators, and need not be particularly described. All the moving parts are designed to secure sufficient strength with the utmost lightness, by which the effect of inertia and momentum is reduced to the least possible amount.

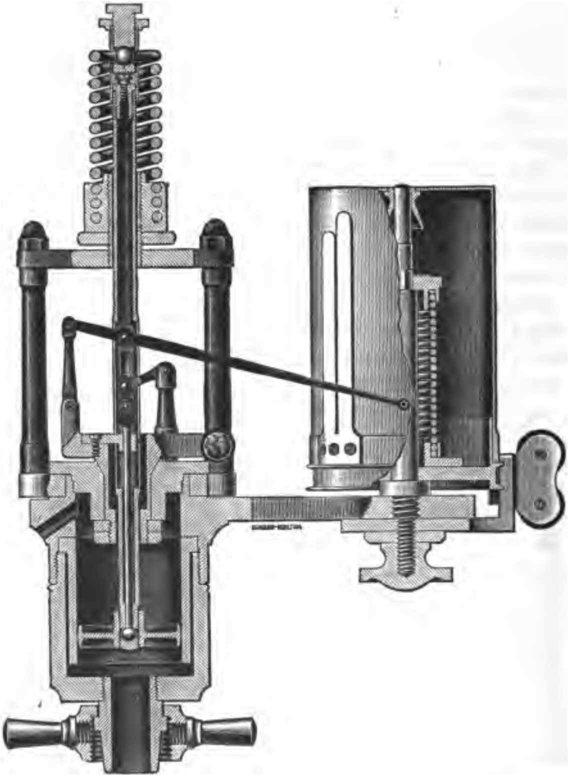
All Crosby Indicators which are numbered above 3.737 are changeable from right-hand to left-hand instruments if occasion requires.

From the design of the Crosby Indicator as above set forth, — the conformation and purpose of its several parts, — it will be seen that every opportunity to improve the instrument has been taken; add to this the fact that only the most skilful workmen of long training in the art are employed, and that every part is made to a standard size by modern specialized machinery, with tools perfectly adapted to their work, and it will be admitted that the proper means have been taken to produce a first-class indicator. We believe this object has been accomplished, and that in competent hands the Crosby Indicator is capable of yielding more accurate results than have hitherto been attainable with any other instrument.

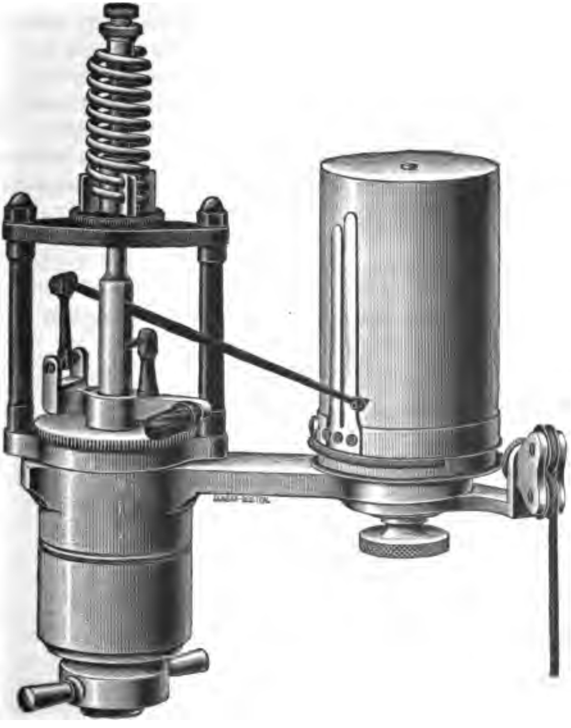
The Crosby Indicator is ordinarily made with a drum one and one-half inches in diameter, this being the correct size for high speed work, and answering equally well for low speeds. If, however, the indicator is to be used only for low speeds, and a longer diagram is preferred, it can be furnished with a two-inch drum.

CROSBY NEW INDICATOR.

Patented.



This instrument is a departure from the ordinary steam engine indicator. One difference is in the location of the spring. This has been removed from the cylindrical case near the piston to the outside and affixed above the moving parts, where it will remain cool under all conditions of use. Whatever error arises from heat, therefore, as affecting the spring in the ordinary indicator, is not present in this instrument. The other and more important difference lies in the size and shape of the piston. This piston is one square inch in area; and is in form the central zone of a sphere. This increased area of the piston provides great active force with a very light pencil mechanism. It is attached by a rod directly to the upper part of the spring, and moves freely and without restraint notwithstanding there may be eccentricity in the action of the spring. In other words, this piston serves as a universal joint to take care of the torsional strains of the spring when it operates the pencil mechanism of the indicator. The pencil mechanism is connected to the piston by means of a rod having at its lower end a ball, which fits into a socket at the piston.



This rod slides through a sleeve attached to the base of the pencil mechanism, and, moving in a vertical line, compels the pencil to move also in a vertical line. Thus, any motion of the piston due to the movements of the spring, which causes the spring rod to deviate, will not affect the pencil mechanism in its vertical course. The contact of the piston with the interior side of the cylinder is a line, and does not induce friction. Ordinarily, the piston of an indicator is a short cylinder fitted to slide easily within another cylinder. Such a piston is usually about one-half inch long, and in use will develop friction throughout its circumference. The piston so made must resist and overcome if possible the eccentricities of the spring in action. Yet, even then, there is always a want of freedom, notwithstanding there are devices to aid the piston in such case. This condition tending to error is recognized by engineers, and considered in the computations made of the diagram taken by the indicator. The freedom of the piston movement in this indicator dispenses with the necessity of this correction. When an indicator is to be used in highly superheated steam or in gases of high temperature this type, with outside spring, will give most accurate results.

This indicator is also made for gas engine work. Price on application.

CHAPTER III.

THE CROSBY GAS ENGINE INDICATOR. THE CROSBY COMBINED GAS AND STEAM ENGINE INDICATOR. THE CROSBY AMMONIA INDICATOR. THE CROSBY ORDNANCE INDICATOR. THE CROSBY HYDRAULIC INDICATOR.

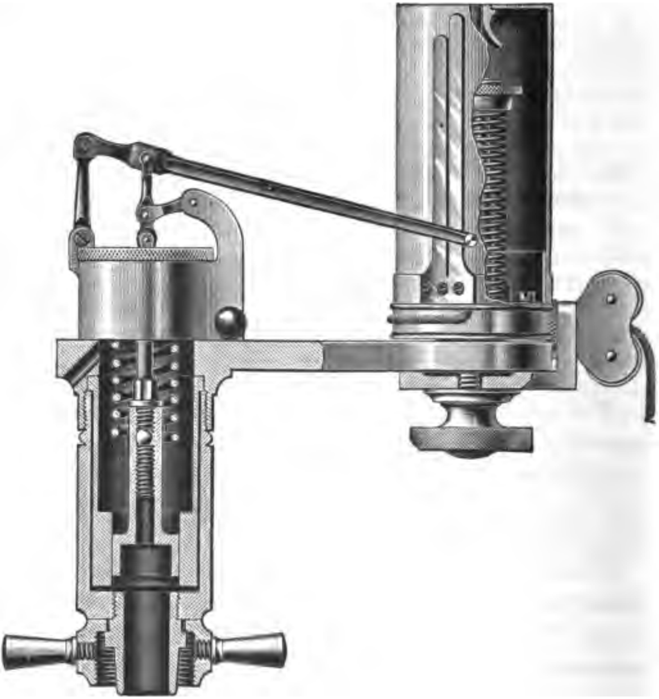
THE GAS ENGINE INDICATOR.

The increase in the use of gas and of gasoline engines has created a demand for an accurate indicator capable of withstanding the heat, the high pressure, and the shock which result from the explosions in the cylinder.

The Crosby Gas Engine Indicator designed to meet these requirements has given perfect satisfaction. The piston is one-fourth of a square inch in area and springs made for one-half inch pistons have their rating doubled when used in this instrument.

CROSBY COMBINED GAS AND STEAM ENGINE INDICATOR.

Patented.



This indicator is for use with either the steam engine or with gas or oil engines, and is supplied with two pistons, either of which can be fitted separately as desired. The piston for steam is one-half of a square inch in area, the same as is used with the steam engine indicator, the one for gas or oil is one-quarter of a square inch in area.

The cylinder is of special construction to suit the respective diameters of the two pistons.

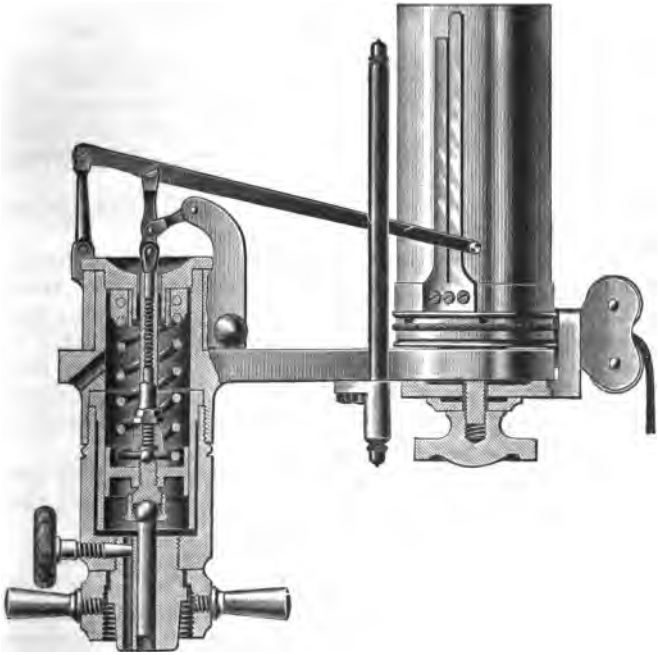
CROSBY AMMONIA INDICATOR.

Patented.

The Crosby Ammonia Indicator is made like the Crosby Steam Engine Indicator, except that all surfaces exposed to the action of ammonia are of steel. For ammonia indicators, both common and three-way cocks are made of steel.

CROSBY ORDNANCE INDICATOR.

Patented.



This instrument will give a true record of high pressures, such as obtain in the operation of the pneumatic gun carriage for heavy ordnance, or in hydraulic pumps.

The pencil mechanism is strong, and has a post bearing lightly against the pencil arm to keep it in contact with the drum during sudden shocks.

The piston is one-fortieth of a square inch in area, and is fitted into a cylinder at the bottom of the instrument. There is a by-pass by which the pressure may be transmitted to the larger piston above, where the pressures to be recorded are not too high for the capacity of the spring. This by-pass is closed by a cock when the small piston is to be used.

CROSBY HYDRAULIC INDICATOR.

Patented.

The Crosby Hydraulic Indicator differs from the one shown in the cut in that it has no by-pass, and in that in place of the piston in the upper chamber a guide is substituted. It is a strong and efficient instrument.

A detent attachment is furnished with the instrument when required. (See cut page 36.)

Every part of the Crosby Indicator is perfectly adapted to its particular function, also to its relation to all the other parts, in size, proportion, and material, and finer results can be obtained with it than with any other instrument. Its smaller size and lighter weight serve to protect it from accident and so contribute to its durability and to the facility with which it can be handled.

Its superior qualities were duly recognized at the Paris Exposition of 1889, where it received the highest award,

A GOLD MEDAL.

Again, at the World's Columbian Exposition, Chicago, in 1893, where it received the highest award,

A MEDAL AND DIPLOMA,

and again, at the Cotton States and International Exposition at Atlanta, in 1895, and at the Russian

Exposition held at Nijni Novgorod, in 1896, where it received the highest award,

A GOLD MEDAL AND DIPLOMA OF HONOR.

In 1904 at St. Louis from the Louisiana Purchase Exposition it received the

GRAND PRIZE.

Its special features are fully protected by Letters Patent in the United States and Great Britain.

The Crosby Indicator is approved and adopted by the U. S. Government. It is the standard in nearly all the great electric light and power stations of the United States. It has been approved and adopted by the principal navies, the government shipyards, and the most eminent technical schools of the world.

Full particulars for the proper care and handling of the Crosby Indicator accompany each instrument.

PRICE.

One nickel-plated Crosby Steam Engine Indicator, with fittings complete, viz.: 1 spring, 1 scale, 2 straight cocks, 1 hollow wrench, 1 screw-driver, 1 bottle of oil, 1 hank of cord, 100 metallic-faced cards, and one copy of "Practical Instructions for using the Indicator," all securely mounted in a velvet-lined mahogany case . . .	$85.00

One three-way cock *instead* of two straight cocks will be furnished if desired.

Address Inquiries and Orders to the

CROSBY STEAM GAGE & VALVE CO.,
93–95 OLIVER STREET, BOSTON, MASS.
16 DEY STREET, NEW YORK, N. Y.
21–23 WEST LAKE STREET, CHICAGO, ILL.
147 QUEEN VICTORIA STREET, LONDON, E. C., ENG.

CHAPTER IV.

INDICATOR ATTACHMENTS AND ACCESSORY APPARATUS NEEDED IN MAKING A TEST ON A LARGE ENGINE.

SARGENT'S ELECTRICAL ATTACHMENT.

For Steam or Gas Engine Indicators.

Patented.

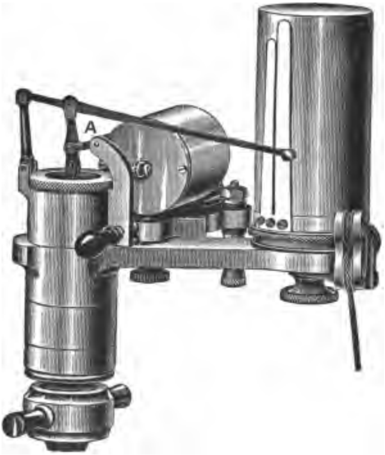


FIG. 3.

In making elaborate tests of power plants, it has heretofore been necessary to employ as many assistants as there were indicators used, but the difficulty of securing simultaneous action on their part is so great that satisfactory work is rarely to be obtained, and more certain means to that end are now considered necessary.

Mr. Frederick Sargent, M. E., invented and patented an electrical device applicable to an indicator, by means of which any number of instruments can be operated and diagrams taken at the same instant of time, simply by closing an electric circuit.

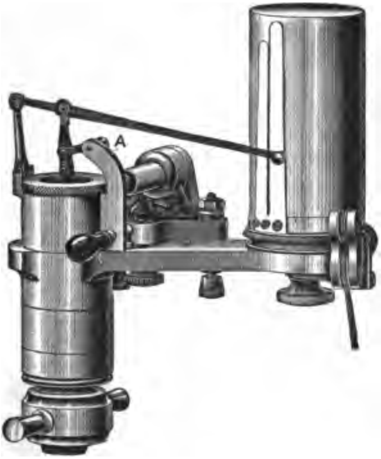


FIG. 4.

We are the sole manufacturers of the Sargent device under United States and English Letters Patent assigned to us. As this invention and these patents are fundamental, in self-protection we are obliged to warn the public against the use of any device which performs a similar duty, as such is an infringement.

Fig. 3 shows a Crosby Indicator fitted with a Sargent Electrical Attachment. Fig. 4 shows a Crosby Indicator fitted with a Circuit Closer.

Description.

Fig. 5 represents Sargent's Electrical Attachment, and consists of an electromagnet, A, which is supported by a bracket, B, which also secures it to the indicator plate. Binding posts, C, C, are attached to the same bracket. The armature D is opposed to the magnet by a spiral spring in the center of the coil, the tension of which is adjustable by means of the screw E, at the back of the magnet. The movement of the armature outwardly is limited by two screws, 1 and 2. To the armature is secured a small latch or hook, F, which is free to work vertically, and engage with a screw-eye inserted in the arm A, Fig. 3. The thumb-screw G is for fastening the attachment to the plate of an indicator through a hole therein.

Fig. 6 represents the Circuit Closer, and is designed to operate the electrically connected indicators, by closing the circuit through them, when the stylus or marking point is put against the paper on the drum of the indicator; to which it is attached. This enables the engineer making the test to control one of the indicators directly by hand — a feature often desirable — and by its use one Sargent attachment is dispensed with.

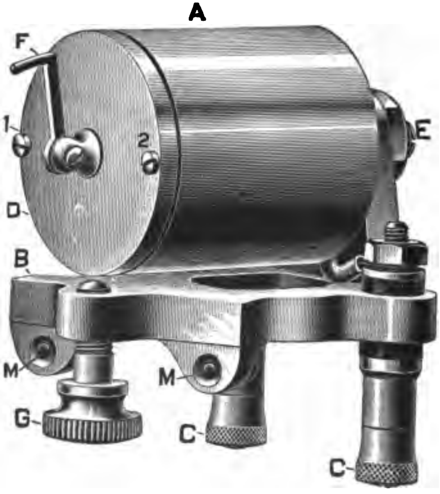


FIG. 5.

It consists of a bracket, H, with a tubular projection, I, fastened to it which contains the circuit closing mechanism. It is attached to the indicator plate by the thumb-screw J, in precisely the same way that the magnets are to the other indicators, and is electrically connected in the same manner through the binding-posts K, K.

To Attach Sargent's Electrical Attachment.

To get the position of the hole in the frame of the indicator, take out the screw G (Fig. 5), and place the bracket holding the magnet against the plate of the indicator, so that the hook F (Fig. 5), when placed horizontally, will point to the middle of the arm A (Fig 3); then scribe through the screw-hole its location upon the plate of the indicator; remove the attachment, and drill a hole where marked that will allow the screw G (Fig. 5) to pass through it. Screw the attachment to the plate and adjust it so that there will be no looseness, turning by the screws M M (Figs. 5 and 6), setting them up gently.

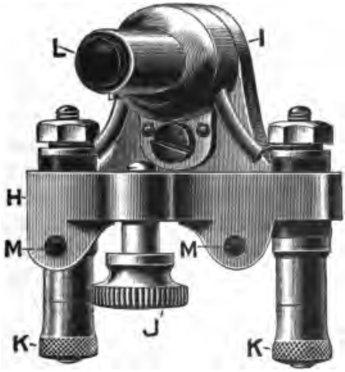


FIG. 6.

Drop the hook F (Fig. 5) to a horizontal position, and bring the arm A (Fig. 3) up to its working position, and mark on it the center of the hole to be drilled for the screw-eye. This hole should be so drilled that the latch will stand level with the plate when in use. The size of the hole may be determined from the screw-eye furnished.

To Attach the Circuit Closer.

The position of the hole in the indicator plate for attaching the Circuit Closer, Fig. 6, is determined in the same manner as for the electromagnet, taking care that the button L (Fig. 6) impinges the center of the arm A (Fig. 3) when the sleeve is turned into the correct position for use.

The sleeve handle of the indicator is unscrewed far enough to allow the button L (Fig. 6) in the end of the projection I to go in as far as it will, then the marking point must be adjusted until it makes the desired tracing on the paper.

To Operate Sargent's Electrical Attachment and Circuit Closer.

For the purpose of illustrating the manner of operating the attachment, assume that it is desirable to procure simultaneous diagrams from a compound engine, taking cards from the ends of each cylinder. Attach the indicators to the engine and arrange the drum motion in the usual manner. On each indicator secure the electrical attachment to its plate by means of screw G, as above described. Make the connections with the battery, having all of the several magnets and the circuit closer in series. Place the paper upon the drum, and bring the pencil arm into such a position as will allow the latch F to drop into the screw-eye before mentioned.

Press the armature firmly against the magnet, and adjust the marking point to the paper in the usual manner. The sleeve handle must be unscrewed enough to allow the full operation of the armature. The circuit should be closed and the armature tension springs adjusted, so that the connected attachments will work simultaneously. Everything should now be in readiness to take diagrams. Connect the drum motions, open the indicator cocks, and as soon as desirable close the circuit, and instantly all of the pencils will be brought against the papers and will remain there as long as the circuit is kept closed.

In order to put on new papers, disengage the drum motions, lift the latch, and swing the pencil arm out of the way.

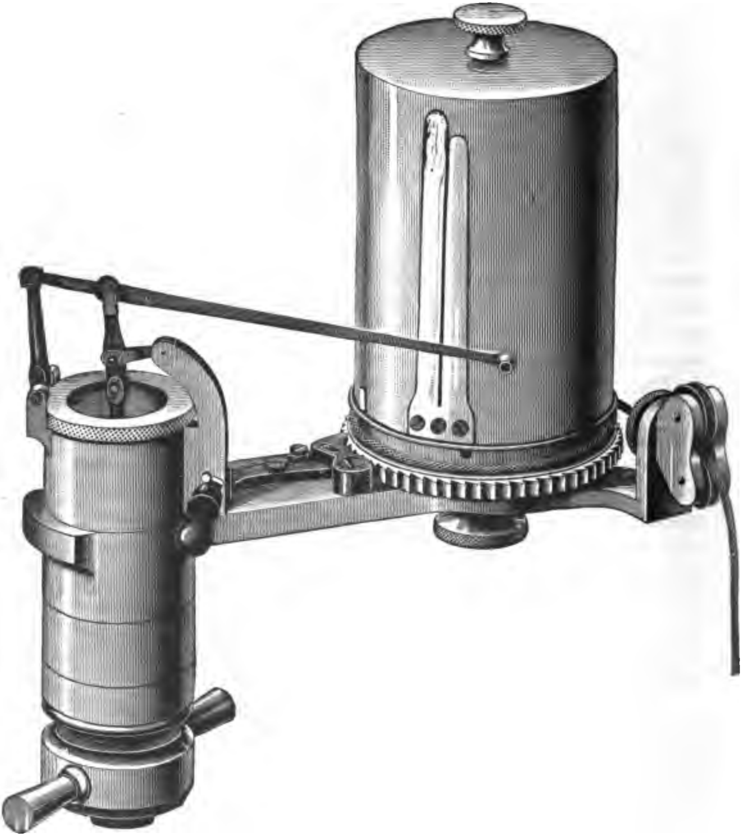
The Electric Battery.

The amount of battery power required will vary with circumstances, and will range from one to two or more cells of a No. 2 Sampson battery, or its equivalent.

The battery for operating the attachments is enclosed in a neat hardwood box with a suitable handle for carrying it, and is sealed so as to prevent slopping. It is very compact and portable, being at the same time extremely active, long lived, and especially adapted to open circuit work.

The connections to the indicator attachments can be made with the battery without opening the box, the binding-posts being on the outside.

This battery, with a quantity of suitable wire for making connections, is furnished with the attachment.

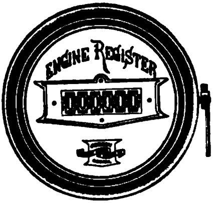


DETENT ATTACHMENT.

The above cut shows the Crosby Steam Engine Indicator provided with a detent attachment and a drum two inches in diameter.

REVOLUTION COUNTER OR ENGINE REGISTER.

Patented.



Engine Revolution Counters, as commonly designed and constructed, depend upon an escapement for receiving the actuating force, and a star-toothed wheel for transmitting the movement to the figure-wheels. The escapement principle, while well adapted to a delicate machine actuated by a constant force, like a watch or clock, is ill-suited to a counter for registering the stroke of a steam engine or other ponderous machine, where the actuating force may be out of all suitable proportion to the structural strength of the counter, and such as to destroy it, if from any cause, like the varying stroke of a pump, the pallet fails to properly engage with the tooth of the wheel. From the same causes such counters may also fail to register. These facts are well known to all mechanics who have had experience with escapement counters. To forcibly illustrate how poorly adapted to a counter the escapement principle is, it is only necessary to state, that in a watch, by the slight force of the actuating spring or weight, it simply *permits* a tooth to *escape*, while in a counter it is required to *move the whole mechanism*, and to do this intermittently and with the varying resistance of one or all of the figure-wheels.

In a Revolution Counter which shall be *reliable*, *durable*, and free from liability to serious injury, the actuating force must, through proper mechanism, be transmitted *directly* and with *certainty* to the figure-wheels, and this can best be done by means of a crank. It matters not in the Crosby Counter whether the movement of the crank is rotary or merely oscillatory, it will count just the same.

The Crosby Improved Revolution Counter, only, employs the crank principle, applied through other simple mechanical motions, so as to record with certainty the operations of any machine, and at the same time obviate all danger of injury to the counter itself or the machine to which it is attached.

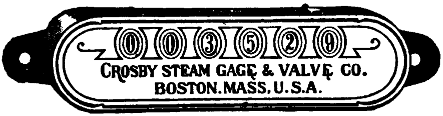
Directions for Attaching the Counter.

The actuating motion of the engine or other machine to which the counter is to be attached, is generally communicated by a rod or bar moving in the same general direction of its length, and the lever should be connected to such rod at a right angle when such rod is in the *middle of its movement.* It should not be clamped rigidly to the shaft until the latter is turned so as to bring the pawl to the *middle of its stroke.* It may be determined, practically, by opening the lid of the counter and watching the movement of the pawl as the shaft is rotated, when the middle point of its travel can be easily fixed. When in this position, clamp the crank to the shaft by means of the set screw.

This arrangement provides for the utilization *of all* the movement of the actuating rod and its greatest effectiveness in moving the mechanism of the counter; and if for any reason the movement of the rod is shorter than its longest possible stroke, — as might happen in the case of a direct acting pump, — there would still be ample motion to insure a correct count.

This counter is adapted to either right or left rotary or reciprocating motions, and is capable of 500 revolutions per minute with safety to the machine and accuracy in the enumeration.

The shaft through which the actuating force is applied may extend from the counter either on the right hand or left hand side as desired.


0 0 3 5 2 9
CROSBY STEAM GAGE & VALVE CO.
BOSTON. MASS. U.S.A.

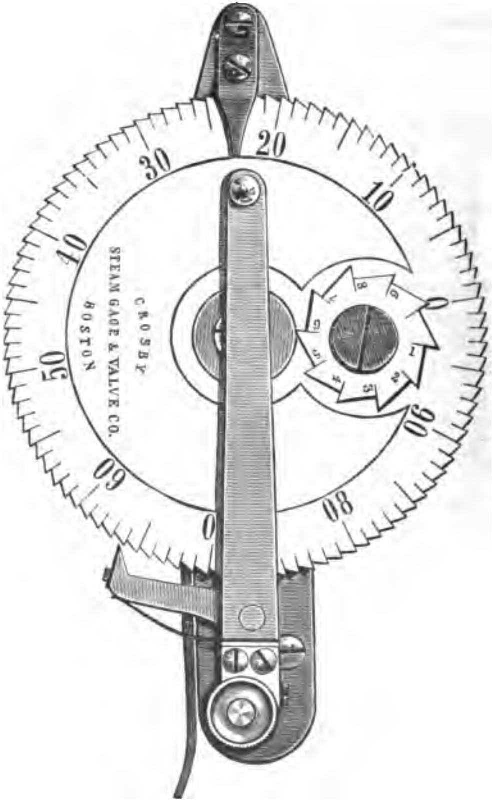
When it is required for rotary motion, it should be stated in the order whether it is to be used for right-handed or left-handed rotation. It is a useful and strong instrument.

This counter is made in the following sizes: —

Size.	
4½ × 1¾ in.	4 Figures
5 × 1¾ ,,	5 ,,
5½ × 1¾ ,,	6 ,,
7 × 2½ ,,	4 ,,
8 × 2½ ,,	5 ,,
9 × 2½ ,,	6 ,,
10 × 2½ ,,	7 ,,

LOCOMOTIVE COUNTER.

For High Rotative Speeds.

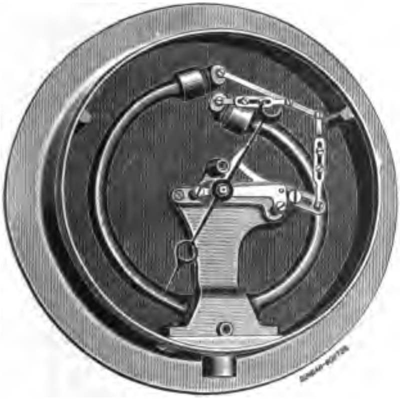


The above cut illustrates our Locomotive Counter. It is designed particularly for use on locomotives and high-speed engines, and is a valuable auxiliary to the steam engine indicator. The arm which moves the ratchet is connected by a cord with some reciprocating part of the engine, or with the drum motion, so as to give it about 1½ inches swing back and forth during each revolution of the shaft. It is provided with a convenient starting and stopping device, so that it can be made to begin or stop counting at any instant.

Recording counters capable of registering speeds as high as 4,000 revolutions per minute are now made by this Company.

PRESSURE AND VACUUM GAGES.

Important.



Accuracy being the essential feature in all gages, whether pressure or vacuum, we take pleasure in informing all who use our gages, that every effort is made to insure this important condition.

The method by which our gages are tested and graduated, is one which insures a truthful and reliable gage. Each one is subjected to a pressure which is given by accurately measured or standard weights, and is graduated to that pressure and not by comparison only with another gage.

An equally accurate method is used in the test-

ing of vacuum gages; each one is tried, measured, and adjusted by an apparatus in which is produced the absolute condition of a vacuum.

The device and process by which this is done is controlled by us.

A Truthful Gage is the Only Good Gage.

Every gage used to ascertain the pressure of steam should have a siphon or some other device, which will furnish to and fill the tube springs with water to keep them cool.

On the back of each gage there is a special notice as to the use of a siphon and the making of repairs.

THERMOMETERS.

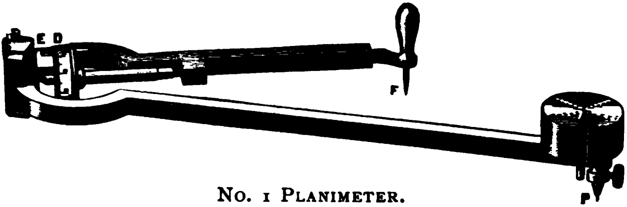
Steam Pipe Thermometer.

Hot Well Thermometer.

Hot Water Thermometer.

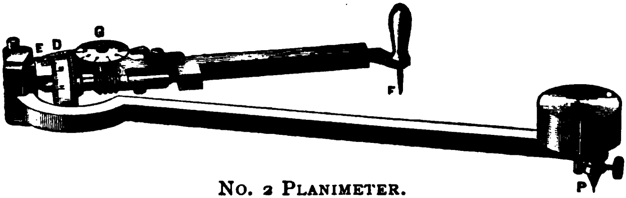
PLANIMETERS.

With Directions for Using the Planimeter on Indicator Diagrams.



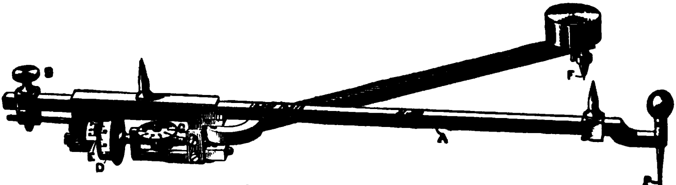
No. 1 Planimeter.

This cut represents the No. 1 planimeter. It is the simplest form of the instrument, having but one wheel, and is designed to measure areas in square inches and decimals of a square inch. The figures on the roller wheel D represent *units*, the graduations on the wheel represent *tenths*, and the vernier gives the *hundredths*.



No. 2 Planimeter.

This cut represents the No. 2 planimeter, which is the same as the No. 1, with the addition of a counting disc G, the figures on which represent *tens* and mark complete revolutions of the roller wheel. By this means areas greater than ten square inches can be measured with facility. The result is given in square inches and decimals, and the reading from the roller wheel and vernier is the same as with No. 1.



No. 3 Planimeter.

This cut represents the No. 3 planimeter, which differs somewhat in design from the two previously described. It is capable of measuring larger areas, and by means of the adjustable arm A, giving the results in various denominations of value, such as square decimeters, square feet, and square inches; also of giving the average height of an indicator diagram in fortieths of an inch, which makes it a very useful instrument in connection with indicator work.

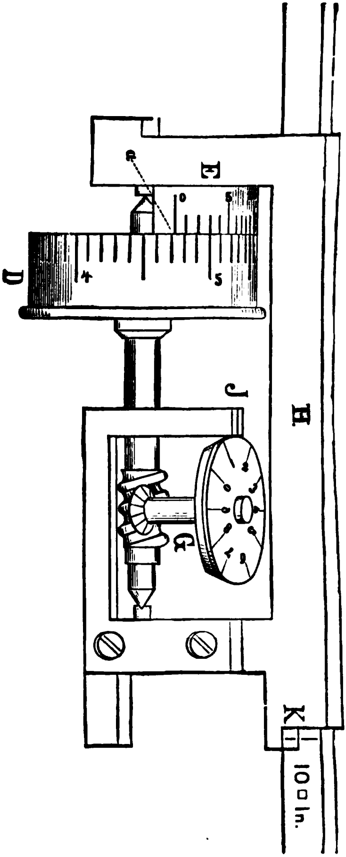


FIG. 7.

RECORDING MECHANISM.

Fig. 7 shows in detail the recording mechanism of a No. 3 planimeter, from which the method of reading from either instrument may be easily understood. G is the counting disc, D the roller wheel, and E the vernier. From the counting disc we read 1 (ten), for the last figure that has passed the index line on the post J; from the roller wheel we read 4 (units), for the last figure that has passed zero on the vernier; we also read 7 (tenths), for that number of graduations beyond 4 that have also passed zero on the vernier (shown by the dotted line *a*), then from the vernier we read 3 (hundredths), because the *third* graduation on the vernier *coincides* with a graduation on the roller wheel.

The complete reading will then be 14.73 square inches.

When starting from zero the movement of the counting disc need not be noted when measuring *single* indicator diagrams, as they are of less than ten square inches area.

Directions for Measuring an Indicator Diagram with a No. 1 or No. 2 Planimeter.

Care should be taken to have a flat, even, unglazed surface for the roller wheel to travel upon. A sheet of dull finished cardboard serves the purpose very well.

Set the weight in position on the pivot end of the

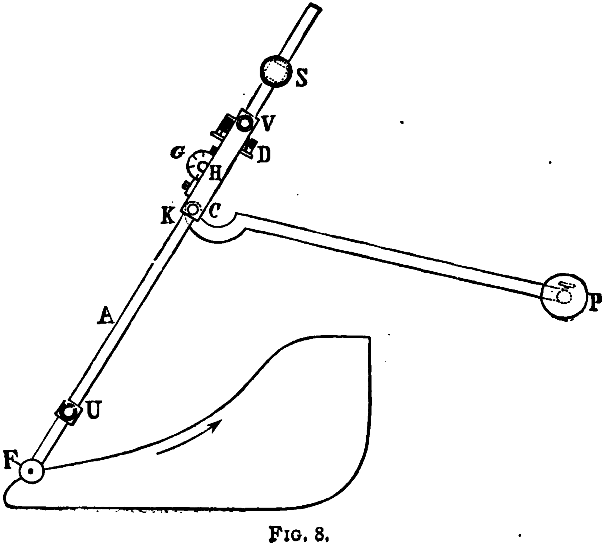


FIG. 8.

bar P, and after placing the instrument and the diagram in about the position shown in the cut (Fig. 8), press down the needle point so that it will hold its place; set the tracer point at any given point in the outline of the diagram, as at F, and adjust the roller wheel to zero. Now follow the outline of the diagram carefully with the tracer point, moving it in the direction indicated by the arrow, or that of the hands of a watch, until it returns to the point of beginning. The result may then be read as follows: Suppose we find that the largest figure on the roller wheel D, that has passed by zero on the vernier E, to be 2 (units), and the number of graduations that have also passed zero on the vernier to be 4 (tenths), and the number of the graduation on the vernier which exactly coincides with the graduation on the wheel to be 8 (hundredths), then we have 2.48 square inches as the area of the diagram. Divide this by the length of the diagram, which we will call 3 inches, and we have .8266 inches as the *average height* of the diagram. Multiply this by the scale of the spring used in taking the diagram, which in this case is 40, and we have 33.06 pounds as the *mean effective pressure* per square inch on the piston of the engine.

Directions for Using the No. 3 Planimeter.

No. 3 planimeter is somewhat differently manipulated, although the same general principle pertains. The figures on the wheels may represent different quantities and values according to the particular adjustment of the sliding arm A. If it is desired merely to find the area in square inches of an indicator diagram, set the sliding arm so that the 10□ inch mark will exactly coincide with the vertical mark on the inner end of the sleeve H at K, Fig. 7. The sliding arm is released or made fast by means of the set-screw S.

With the wheels at zero and the planimeter and diagram in the proper position (shown in Fig. 8) trace the outline carefully, and read the result from the roller wheel and vernier, the same as directed for the No. 1 and No. 2 instruments.

Example: Suppose, in a diagram so measured, we read from the figures on the roller wheel 3 (units), from the graduations on the roller wheel 0 (tenths), and from the vernier 8 (hundredths); then we have an area of 3.08 square inches; to find the average height we divide this by its length, which is 3½ inches. Thus, 3.08 ÷ 3.5 = .88 inches the average height. This multiplied by the scale of the spring used, which in this case is 60 lbs., and we have 52.8 lbs. as the M. E. P.

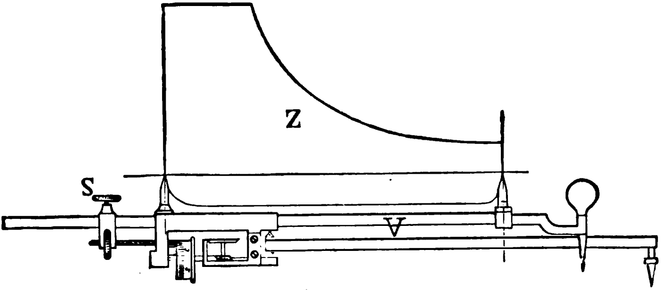


FIG. 9.

To find the average height of an indicator diagram at one measurement, set the sliding arm so that the steel points on its upper side shall be just the length of the diagram — measured on a line parallel with the atmospheric line — apart, as shown in Fig. 9. With this adjustment the figures on the counting disc represent hundreds, those on the roller wheel tens, the intermediate graduations units, and the vernier gives the decimal. Place the instrument in position with the wheels at zero, and trace the outline of the diagram. The result of the reading will be its *average height* in fortieths of an inch.

Example: Measuring the same diagram as before, suppose we read from the figures on the roller wheel 3 (tens), from the graduations 5 (units), and from the vernier 2 (tenths); then we have 35.2 *fortieths* of an inch, which, divided by 40, gives .88 of an inch, the average height. This multiplied by the scale of the spring used and we have 52.8 lbs. M. E. P., the same as in the last example. A simpler process is to multiply the reading by the *factor* corresponding with the scale of the spring, which for a 60 lb. spring is 1.5, then we have $35.2 \times 1.5 = 52.8$ lbs., the same as by the other process.

Following is a list of the pressures or scales to which indicator springs are commonly made, with their corresponding *factors* immediately below.

Springs,	8	12	16	20	24	30	40	50	60	80	100	120	150	180
Factors,	0.2	0.3	0.4	0.5	0.6	0.75	1.0	1.25	1.5	2.0	2.5	3.0	3.75	4.5

When two diagrams are taken on the same card they may be measured conjointly and the average height divided by two to get the average height of both. We, however, recommend that each diagram be measured separately, especially if there is a difference in their areas, which is generally the case.

When there is a loop in the diagram caused by the steam expanding below the back pressure line when the engine is non-condensing, its outline should be traced in the same way as directed for a plain diagram, as the principle on which the planimeter works is such that the area of the loop will be subtracted from the main part of the diagram, and the reading of the instrument when the measurement is completed will be the correct net area sought.

When one has become familiar with the use of the planimeter it is not necessary always to set the wheels at zero as required in the foregoing directions, but their reading as they stand just before beginning to trace a diagram may be noted down and this quantity subtracted from the reading when the tracing is completed. The difference between the two readings is the area sought.

For instance: suppose we find that the reading of the wheels, including the counting disc, at the beginning is 47.31, and when the tracing is completed it is 49.43, then 49.43 — 47.31 = 2.12 square inches, the area measured. Then to measure a second diagram, note down the last reading, viz.: 49.43, and when the tracing is completed we read 51.63. Then 51.63 — 49.43 = 2.20 square inches, the area of the second diagram.

The foregoing directions for using the planimeter are applicable to any single diagram.

The use of Amsler's Polar Planimeter in the measurement of indicator diagrams enables one to measure ten cards with it in the time which would be required to measure one card by any other method, and it insures the utmost accuracy in the work.

The planimeter is a precise and delicate instrument and should be handled and kept with great care, in order that it may be depended upon to give correct results. After using, it should be wiped clean with a piece of soft chamois skin.

THE THROTTLING CALORIMETER.

In order that the test of an engine or boiler may be complete a determination should be made of the quality of the steam, *i.e.*, the priming or the amount of moisture carried by the steam. This determination was formerly made by methods which could be made to give satisfactory results in the hands of a physicist or a trained expert, but which were troublesome and unreliable when employed by an inexperienced observer. The quality of steam delivered by a boiler or supplied to an engine can now be determined with ease and certainty by aid of the throttling calorimeter, invented by Prof. C. H. Peabody of the Massachusetts Institute of Technology, and described by him in the * "Journal" of the Franklin Institute, and in the † "Proceedings of the American Society of Mechanical Engineers."

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* "Journal" Franklin Institute. June, 1888. Volume CXXVI., Page 134.

† "Proceedings American Society of Mechanical Engineers," 1888-89, Volume X., Page 327, and 1889-90, Volume XI., Page 193.

The throttling calorimeter depends on the principle that steam which contains a moderate amount of moisture will become superheated if the pressure is reduced by throttling, without loss of heat. The form here shown is simple, substantial, and inexpensive, and has been used by the inventor and others

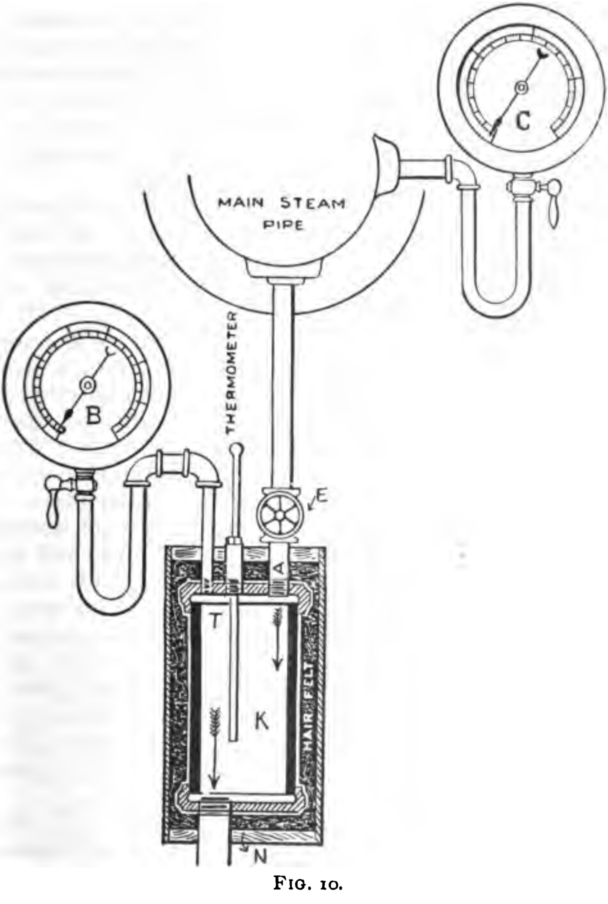

FIG. 10.

with complete satisfaction. The calorimeter, shown in Fig. 10, is a closed cylindrical metallic chamber K, having an inlet passage at A, controlled by the valve E, an outlet passage at the bottom N, and a thermometer cup at T. The chamber is thickly wrapped with asbestos and hair felt, protected by wood lagging to reduce radiation and loss of heat. The U shaped tubes or siphons for attaching the pressure gages B and C are furnished with the calorimeter; the gages and thermometer are extra, and may be furnished or not, as required.

The nipple A, connecting the inlet valve E with the chamber K, is made of composition, cut with pipe thread and provided with a well-rounded orifice for gaging the flow of steam as shown by the full size Fig. 11.

The connection with the main steam pipe from which a sample of steam to be tested is taken, should be as short and direct as possible, and should be well wrapped to reduce radiation. The supply pipe should enter the main steam pipe at least half an inch, or it may be long enough to reach across the bore of the steam pipe, and be pierced with numerous small holes and closed at the end. The waste pipe N should be at least one inch in diameter for its entire length, and may be larger if longer than twenty feet. The gage C for measuring the pressure in the main steam pipe must be attached directly to that pipe close to the calorimeter.

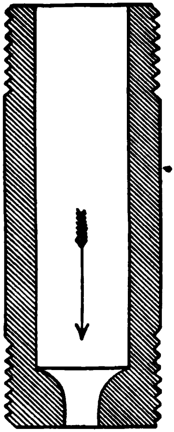

FIG. 11.

To use the calorimeter, fill the thermometer cup with oil and insert the thermometer; see that the siphons are filled with cold water and that they do not leak; open the valve E wide, and wait ten or fifteen minutes till the whole apparatus is heated. Read the gage B and add the pressure of the *atmosphere to get the absolute pressure in the calorimeter; find the corresponding temperature from a table of the properties of saturated steam and compare with the temperature in the calorimeter given by the thermometer; the excess of the latter over the former is the superheating of the steam in the calorimeter. The flow of steam through the calorimeter will be sufficient to make the loss by radiation of no consequence and no correction need be applied.

When all is ready, read the pressure of the steam p in the main steam pipe, the temperature t_s in the calorimeter, the pressure $p_{,}$ in the calorimeter, and take the pressure p_a of the atmosphere. From a table of the properties of saturated steam, find the temperature $t_{,}$ corresponding to the absolute pressure $P_{,} = p_{,} + p_a$.

From the same tables find the total heat $\lambda_{,}$ corresponding to the pressure $P_{,}$; also the heat of vaporization r and the heat of the liquid q corresponding to the absolute pressure in the steam pipe

$P = p + p_a$.

The weight or moisture in one pound of moist steam drawn from the steam pipe is to be calculated by the equation

$$\text{Priming} = 1 - \frac{\lambda_{,} + 0.48\,(t_s - t_{,}) - q}{r}$$

in which the factor 0.48 is the specific heat of superheated steam at constant pressure.

The calculation will be readily understood from the following example:

Pressure in steam pipe, $p = 69.8$ pounds.
Pressure in the calorimeter $p_{,} = 12$ pounds.
Pressure of the atmosphere, $p_a = 14.8$ pounds.
Temperature in the calorimeter, $t_s = 268.2°$ F.
Absolute pressure in steam pipe, $P = p + p_a = 69.8 + 14.8 = 84.6$ pounds.

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*Note.—The pressure of the atmosphere is commonly assumed to be 14.7 pounds per square inch; it may be taken by aid of a barometer or obtained from published records of the Weather Bureau for the day. Inches of mercury can be reduced to pounds per square inch by multiplying by 0.49.

Absolute pressure in calorimeter, $P_{,} = p_{,} + p_a = 12 + 14.8 = 26.8$ pounds.

Temperature of saturated steam at 26.8 pounds, 243.9° F.

Total heat at 26.8 pounds, $\lambda_{,} = 1156.4$ thermal units.

Heat of vaporization at 84.6 pounds, $r = 892.7$ thermal units.

Heat of the liquid at 84.6 pounds, $q = 285.5$ thermal units.

$$\text{Priming} = 1 - \frac{1156.4 + 0.48\ (268.2 - 243.9) - 285.5}{892.7} = 0.012:$$

a result that is commonly stated as 1.2 per cent. priming.

Steam delivered by a boiler or supplied to an engine commonly contains a small amount of moisture, but if the steam is very wet from any cause it may fail to superheat in the calorimeter and in such case the calorimeter cannot be used for determining its quality. Should this occur in a boiler test, it indicates either that the design of the boiler is defective or that it is in bad condition and needs cleaning. The steam supplied to an engine may be deprived of the greater part of its moisture, if it be very wet, by passing it through a separator. It has been found that steam used in good ordinary practice will always superheat in the throttling calorimeter.

While it is advisable that the gages and thermometer used with the throttling calorimeter should be of first quality and entirely reliable, the errors that such instruments are liable to have do not have a serious effect on the result. Thus, at 100 pounds absolute and with atmospheric pressure in the calorimeter, 10° F. superheating indicates 0.035 priming; should the thermometer be wrong 5° F. and indicate 15° F., the priming will appear to be 0.032. In a similar manner it will be found that an error of a pound or two in the pressure of the steam in the steam pipe will have an insignificant effect on the result of a test. The effect of an error in the reading of the pressure in the calorimeter is somewhat more serious and care should be taken to have that gage correct.

A glass U tube filled partially full of mercury is sometimes used to give the pressure in the calorimeter instead of the gage B.

The leg of this column which connects with the calorimeter will after a short time have some water collect on top of the mercury.

Water should be poured into the other or open leg of the column so as to keep the amount in each leg the same.

CHAPTER V.

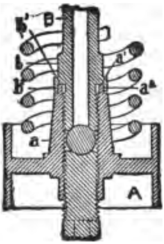
HOW TO HANDLE AND TAKE CARE OF A CROSBY INDICATOR.

The indicator is a delicate instrument, and in order to secure good results from its use, it must be handled with care and be kept in good order.

To remove the piston and spring, unscrew the cap; then take hold of the sleeve and lift all the connected parts free from the cylinder. This gives access to all the parts to clean and oil them.

Never remove the pins or screws from the joints of the pencil movement, but keep them well oiled.

Important. In the under side of the shoulder of the piston-rod B, is a circular channel formed to receive the upper edge of the slotted socket of the piston A. Whenever it is desirable to connect the piston-rod with the piston, either in the process of attaching a spring, or for the purpose of testing the freedom of movement of the piston in the cylinder without a spring, BE SURE *to screw the piston-rod into the socket as far as it will go;* that is, until the upper end of the socket, a^1, is brought firmly against the bottom of the channel, b^2, in the piston-rod. This insures a perfectly central alignment of the parts and therefore a perfectly free movement of the piston within the cylinder.



To attach a Spring. Hold the hollow wrench in an inverted position and insert the piston-rod until its hexagonal part engages the wrench; then, with the spring inverted, insert the combined wrench and piston-rod until the bead of the spring rests in the concaved end of the latter; then invert the pis-n and pass the transverse wire at the bottom of the spring through the slot until the threads at the bottom of the piston-rod engage those inside the socket of the piston, and with the wrench screw it in as far as it will go; that is, until the upper edge of the socket is in contact with the bottom of the channel in the shoulder of the piston-rod. The piston screw should be loosened slightly before this last operation and afterwards set up against the bead lightly, to provide against any lost motion, yet not so as to make it rigid. Next, hold the sleeve and cap in an upright position, — so that the pencil lever will drop to its lowest point, — and engage the threads of the swivel head with those inside the piston-rod and screw it up until the threads on the lower projection of the cap engage those in the spring head, and continue the process until the latter is screwed firmly up against the cap. Then, letting the cap go free and holding only by the sleeve, continue to turn the piston (together with its connections) until the top of the piston-rod is flush with the shoulder on the swivel head.

The piston and its connections may now be inserted in the cylinder and the cap screwed down, which will carry all parts into their proper places.

To detach a spring simply reverse this process.

To change the location of the atmospheric line of the diagram. First, unscrew the cap and lift the sleeve, with its connections, from the cylinder; then, — holding the sleeve with the left hand, — with the right hand turn the piston and connected parts towards the left, and the pencil point will be raised, or to the right and it will be lowered. One complete revolution of the piston will raise or lower the pencil point ⅛ inch and this should be the guide for whatever amount of elevation or depression of the atmospheric line is needed.

To change to a left-hand instrument. If it is desired to make this change: First, remove the drum by a straight upward pull; then, with the hollow wrench remove the hexagonal stop screw in the drum base, and screw it into the vacant hole marked

L; next, *reverse* the position of the adjusting handle in the arm; also, the position of the metallic point in the pencil lever; then replace the drum and the change from right to left will be completed.

This applies to all indicators numbered above 3737.

The tension on the drum spring may be increased or diminished according to the speed of the engine on which the instrument is to be used, as follows: Remove the drum by a straight upward pull; then raise the *head* of the spring *above* the *square* part of the spindle and turn it to the right for more or to the left for less tension, as required; then replace the head on the spindle.

Before attaching the indicator to an engine, be sure to *blow steam freely through pipes and cock* to remove any particles of dust or grit that may have lodged in them.

After using the indicator it should be carefully wiped and oiled.

For this purpose it is not often necessary to disturb the paper drum, but the cylinder cap should be unscrewed and all the connected parts lifted out; then the piston, piston-rod and spring should be detached and all carefully wiped with cloth or tissue paper until perfectly dry, then slightly oiled with a lubricant of good quality; the inside of the cylinder should also be oiled. After this is done the piston and piston-rod should be replaced in the cylinder, but the spring should be kept on its stud in the box when the instrument is not in use. This box should be kept in a dry place, locked, and the key in the owner's pocket.

After the indicator has laid unused for any length of time the oil used at its last cleaning may have become gritty or gummy, and should be wiped off with a soft cloth or tissue paper saturated with naphtha or benzine, and freshly oiled before using it again. This keeps the instrument in prime condition and insures the best results from its use. An occasional naphtha bath is good for an indicator, as it thoroughly cleanses every part.

If any grit or other obstruction gets into the cylinder it will seriously affect the diagram and lead to bad results. It is not difficult to detect such trouble and it should be remedied at once by taking out the piston, detaching the parts and cleaning them as above described, when the disturbing cause will generally be removed.

It is essential to know whether or not the indicator is in good condition for use; especially, to know that the piston has perfect freedom of motion and is unobstructed by undue friction. To test this successfully, detach the spring and afterwards replace the piston and piston-rod in their usual position, then, holding the indicator in an upright position by the cylinder in the left hand, raise the pencil arm to its highest point with the right hand and let it drop. It should freely descend to its lowest point. The inner walls of the cylinder should be frequently lubricated. The pencil should always have a smooth fine point; for this purpose a fine file is the best instrument to use.

INDICATOR SPRINGS.

Indicator springs are made of different sizes of steel wire, to adapt them, in point of strength, to different pressures of steam.

To adapt this indicator to any steam pressure, springs are made to the following scales: —

4 8 12 16 20 30 40 50 60 80 100 120 150 180

The number stamped on the spring represents the number of pounds pressure to the square inch which are required to compress it sufficiently to move the pencil vertically one inch on the diagram. The strength of the spring that should be used for indicating an engine depends upon the maximum steam pressure to which it will be subjected; it should be of such a strength or number that the diagram will not be over 1¾ inches high. The proper spring to be used in any given case may be found as follows: Divide the boiler pressure, expressed in pounds, by the desired height of the diagram, expressed in inches, and the result will be the number of the spring required. For instance, if the boiler pressure is 70 pounds and the desired height of the diagram is 1¾ inches, then 70÷1¾ =40, the number of the spring required.

In practice, the best diagram for measuring and interpreting is one in which the length is not more than two times the height, for the reason that the points of cut-off, exhaust and compression are better defined than in a longer diagram.

The late John C. Hoadley, an eminent authority in the use of the steam engine indicator, said:—

"There are good reasons for keeping the diagram of very moderate length. From 2½ to 3 inches will be found long enough to admit of all useful division, and the movement of the paper cylinder will be slower, and the tracing correspondingly more delicate than if a longer card is made. A similar remark applies equally well to the vertical motion, which can be reduced to any amount desired by using springs of suitable stiffness."

Sir Frederick Bramwell succeeded in obtaining very satisfactory diagrams at extraordinary speeds and high pressures, by limiting the dimensions to 1 inch in length and width, but this was an extreme case. It is generally more convenient and satisfactory to make the length about 2½ to 3½ inches, and the height from the atmospheric line about 1½ to 1¾ inches, by selecting a spring adapted to this pressure.

INDICATOR SCALES.

The scales for measuring diagrams are sometimes made of steel, with several different graduations marked on each; they are more often made of boxwood, and these we recommend as being more easily read and less liable to be misapplied, as there is only one graduation marked on each scale of this kind. The scale with which to measure the height of a diagram must correspond with the number of the spring used in the indicator when the diagram is taken. For a diagram taken with a number 40 spring, use a scale graduated in 40 divisions to the inch, or for a diagram taken with a number 60 spring, use a scale graduated in 60 divisions to the inch, and so of all other springs.

Place the scale on the diagram at right angles to the atmospheric line with the zero mark of the scale exactly on that line; the figures set against these divisions show — at whatever point the line of the diagram crosses the edge of the scale — the pressure per square inch exerted by the steam on the piston of the indicator in tracing it. The divisions below the zero point show vacuum.

The most common scales are those numbered 40, 50, and 60; that is, an inch of vertical height on the scale represents, according to the number of the spring used, 40, 50, or 60 pounds of steam pressure per square inch in the cylinder.

CHAPTER VI.

HOW AND WHERE TO ATTACH THE INDICATOR.

The indicator should be attached close to the cylinder whenever practicable, especially on high-speed engines. If pipes must be used they should not be smaller than half an inch in diameter and as short and direct as possible; if long pipes are needed they should be slightly larger than half an inch and covered with a non-conducting material.

Diagrams should be taken from both ends of the cylinder of an engine. If the diagram from one end is satisfactory it is not safe to assume that one taken at the other end will be equally so; it is often otherwise, owing to the varying conditions usually found. The lengths of thoroughfares, the points of valve opening and closing, and the lead, are variable and should be carefully adjusted to secure the best results, and this can only be done through the instrumentality of an indicator.

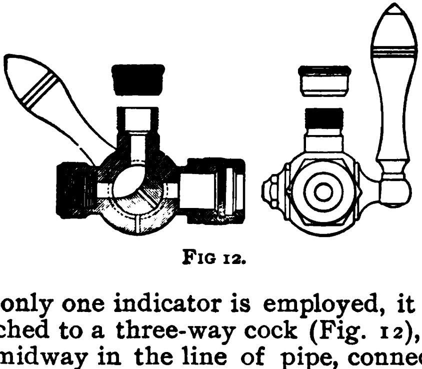


FIG 12.

When only one indicator is employed, it is generally attached to a three-way cock (Fig. 12), which is located midway in the line of pipe, connecting the holes at either end of the cylinder; by this arrangement diagrams can be taken from either end simply by turning the handle of the three-way cock. In such a case, the second diagram should be taken as quickly as possible after the first, so as to be under like conditions of speed, pressure, and load.

The indicator *can* be used in a horizontal position, but it is more convenient to take diagrams when it is in a vertical position, and this can generally be obtained, when attaching to a vertical engine, by using a short pipe with a quarter upward bend. No putty or red lead should be used in making any joints, as particles of it may be carried by the steam into the indicator, and great harm result therefrom; if a screw fits loosely, wind into the threads a little cotton waste, which will make a steam tight joint. The indicator should never be set so as to communicate with thoroughfares where a current of steam will flow past the orifice leading to the indicator, as the diagrams taken under such conditions would be of no practical value.

The cylinders of most modern steam engines are drilled and tapped for the indicator and have plugs screwed into the holes, which can readily be removed and the proper indicator connections inserted. But when this is not the case, the engineer should be competent to do it under the directions here given.

When drilling holes in the cylinder the heads should be removed if convenient, so that one may know the exact position of the piston, the size of ports and passages, and be able to remove every chip or particle of grit which might otherwise do harm in the cylinder or be carried into the indicator and injure it. When the heads cannot be taken off, it can be arranged so that a little steam may be let into the cylinder, when the drill has nearly penetrated its shell, so that the chips may be blown outward, — care being taken not to scald the operator.

Each end of the cylinder should be drilled and tapped for one-half inch pipe thread. The holes must always be drilled into the clearance space, at points beyond the range of the piston when at the end of the stroke, so as not to be obstructed by it, and away from steam passages, to avoid strong currents of steam. By placing the engine on a dead center, it is easy to tell how much clearance there is, and the hole should be drilled into the middle of this space; the same process should be repeated at the other end of the cylinder.

On horizontal engines the most common practice is to drill and tap holes in the *side* of the cylinder at each end and insert short half inch pipes with quarter upward bends, into which the indicator cocks may be screwed; on some horizontal engines it may be more convenient to drill and tap into the top of the cylinder at each end and screw the cocks directly into the holes. On vertical engines, for the upper end of the cylinder the cock may be screwed into the upper *head* or *cover*, and for the lower end, into the *side* of the cylinder, after drilling and tapping the necessary hole. It is preferable to drill the holes in the *sides* of a cylinder rather than the heads, because the former gives better results and requires less pipe and fittings.

Before deciding just where to drill the holes it is wise to consider *all* the conditions of the case and devise the *whole* plan for indicating the engine.

Sometimes a drum motion can be erected more advantageously in one place or position in the engine room than another, or one kind may be better adapted for a given place than another. Again, the type of engine and position of the steam chest, the kind of cross-head and the best means for attaching to it, the position of the eccentric, its rods and connections, all should be taken into account when determining the best places to drill the cylinder and locate the indicator, in order to secure a proper connection with the reducing motion, a perfectly free passage for steam to the indicator and the most convenient access to the instrument for taking diagrams.

CHAPTER VII.

DRUM MOTION.

The motion of the paper drum may be derived from any part of the engine which has a movement coincident with that of the piston. In general practice and in a large majority of cases, the cross-head is chosen as being the most reliable and convenient.

The movement of the cross-head, whatever it actually is, must, by appropriate mechanism, be reduced to the length of the diagram to be taken, and this reduced motion must be, in point of rapidity, in exact ratio to the motion of the piston.

To obtain this reduced motion, various devices may be employed.

The Reducing Lever in some one of its various forms can be easily made and adapted to suit almost any conditions.

A common form of this device is shown in Fig. 13, and answers fairly well for large and quick running engines. It should be made of straight grained pine, one inch or more in thickness, about three inches wide at the top, and tapering to a width of about two inches at the bottom; its length should be at least one and a half times the length of the stroke of the piston.

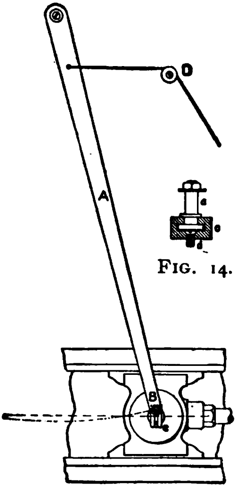


FIG. 14.

FIG. 13.

The lever, A, is suspended by a bolt from the ceiling or from a truss or frame overhead prepared for that purpose, in such a manner as to permit it to swing edgewise and parallel with the guides of the engine. Near the bottom of the lever is a steel stud, secured by a nut on the outside shown at B. This stud has a T-head projecting inwardly from the lever, and is formed to run freely, but without looseness, in a T-slot, cut in an iron plate, and firmly attached to the center of the cross-head, which, as it moves to and fro, gives to the lever the necessary swinging motion.

Fig. 14 shows the arrangement of the T-headed stud, *a*, in connection with the slotted iron plate, *c*; one of the screws by which it is attached to the cross-head is shown at *d*. The head of the stud should be about one inch in diameter and the shank about one-half inch. The T-slot is milled out of a cast-iron plate of suitable size and shape to give the proper run for the stud.

When the lever is perpendicular, or in the middle of its path, the stud should be near the bottom of the slot, which should be long enough to retain the stud when the cross-head is at the extreme end of the stroke.

By this device the bottom end of the lever is moved the full distance that the cross-head travels in either direction, and for this reason it is more accurate than a lever of the same kind having its lower end slotted to work on a stud inserted in the cross-head, as is sometimes used. *D* is a small pulley placed near and on a level with the pin in the lever, for the indicator cord to pass over.*

To find the point on the lever at which to attach the indicator cord, proceed as follows: Divide the length of the lever by the length of the piston stroke, and multiply the quotient by the required length of the diagram, all expressed in inches and decimals of an inch, and the product will be the proper distance from the pivot in the top of lever to the point of attachment.

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* NOTE. — While this form of reducing lever is commonly made with a *pin* for attaching the indicator cord, greater constancy of motion for the drum would be attained by the use of a sector, such as is shown at S in Fig. 15.

For example: If the lever is forty-eight inches long and the piston stroke is thirty inches and we wish to obtain a diagram three and a half inches long, we have $48'' \div 30'' = 1.6''$; $1.6'' \times 3.5'' = 5.6''$, the radius required to give a three and a half inch diagram. If we require a diagram four and a half inches long, then: $1.6' \times 4.5'' = 7.2''$, the radius required to give a four and a half inch diagram.

The object of all mechanisms for actuating the drum of the indicator should be such that the relation of piston to drum movement should be constant. Such constancy cannot, however, be fully attained by the use of any form of reducing lever, and they should not be employed when important adjustments or tests are to be made. Their simplicity and the small expenditure of time and money in their construction may entitle them to favorable consideration on the part of beginners in the use of the indicator, and when only ordinary work is to be done.

The forms shown in Fig. 16, Fig. 17, Fig. 18, and Fig. 19 are correct in principle, and when carefully constructed may be relied upon to give correct results.

The Brumbo Pulley, shown in Fig. 15, is another form of reducing lever, and one often used by engineers, especially on locomotives. It can be quickly and cheaply made and can be used on almost any kind of engine. The swinging lever E is a strip of straight grained pine, one inch or more in thickness, three to four inches

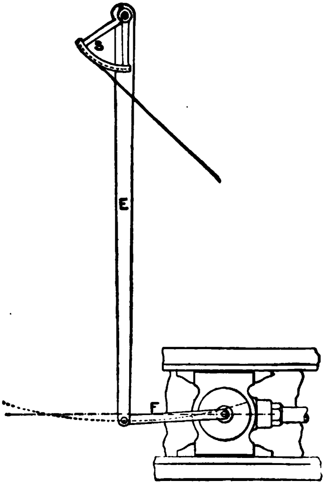


FIG. 15.

wide, and from one and a half to two times as long as the piston stroke. It is suspended by a bolt or screw, serving as a pivot, from a frame or truss overhead constructed for that purpose, and is connected at its lower end by the wooden link F, to the usual stud or pin fixed in the center or other convenient part of the cross-head; the link should be from one-third to one-half the length of the piston stroke. In the illustration the proportions of lever and link are as 60 to 15; the lever being two times and the link one-half the length of stroke.

The sector S may be constructed of wood or of metal as here shown; it has a groove in its circular edge for the cord to run in and is screwed to the upper end of the lever or pendulum, so that its center will coincide with the center of the pivot on which it swings. The radius of the sector which is necessary to give the proper motion to the drum to obtain the desired length of diagram can be found as follows: Divide the length of the lever by the length of the piston stroke and multiply the quotient by the length of diagram desired, and the product will be the required radius, all the terms being expressed in inches and decimals of an inch. For example: If the lever is thirty inches long and the piston stroke is twenty inches, and we wish to obtain a diagram three inches long we have $30'' \div 20'' = 1\frac{1}{2}''$; $1\frac{1}{2}'' \times 3'' = 4\frac{1}{2}''$, the radius required to give a three-inch diagram.

When the conditions are favorable, the lever should be hung so that it will swing in a vertical plane, parallel with the guides and in line with the indicator, as this arrangement is the most simple, and the use of guide pulleys is avoided. It is not absolutely necessary, however, that the lever shall swing in a vertical plane, but it may swing in a plane at any angle thereto, where the conditions require it. In such cases a man's ingenuity and inventive faculty must aid him. A link made of a thin strip of steel, that will twist a little, is in some cases found very convenient.

When the cross-head is at mid-stroke the lever must hang plumb and the pin which connects its lower end to the link must be as much *below* the horizontal line of motion of the *stud* in the cross-head, *as it* sweeps *above* that line at either end of the stroke. See the dotted line for an illustration of this point, which is important. The cord must lead from the sector in about the same plane with its swing.

Carrying pulleys should be avoided as far as possible, but whatever number is necessary should be firmly placed. The swinging arm of the guide-pulley on the indicator should always be fixed in the direction from which the cord is received.

A piece of piano wire is often used to replace the string leading from the sector to the indicator.

Some engines are furnished with a drum motion of this kind, made of steel with nicely fitted joints, which can be readily attached to the engine, and are very convenient to use.

The Pantograph, illustrated in Fig. 16, is another style of reducing motion. Although theoretically it gives a perfect motion, owing to its many joints it may become shaky and give erroneous results, unless it is very nicely made and carefully used. When the indicator is applied to the side of the cylinder the pantograph works in a horizontal plane. The pivot end B rests on a post or other support set opposite to the middle of the guides, and the working end A receives motion from the cross-head, — to which it is attached by a suitable iron with a hole drilled in it for the stud A to work in. By adjusting the support for the pivot end to the proper height and at a proper distance from the guides, the cord may be carried directly from the pin E to the indicator without the need of carrying pulleys. The movable bar may be set forward or backward by the pins D—C, so as to perfectly adjust the movement of the pin E to the required length of the diagram; this pin must always be in a straight line with the stud A and the pivot B. The string from E should always lead off in a line parallel to the piston-rod.

The Reducing Wheel is another device for giving the proper motion to the paper drum. Although old in principal, and as formerly made not highly approved by experienced engineers, this style is now coming into more general use, and the superior manner in which it is designed and constructed, seems to warrant this change.

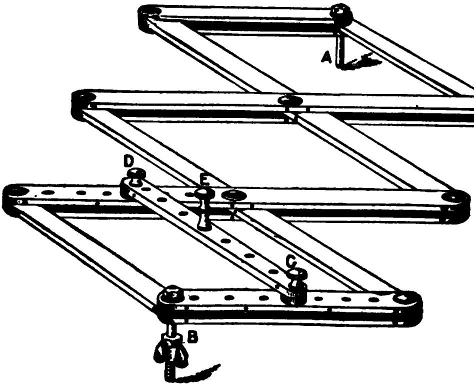
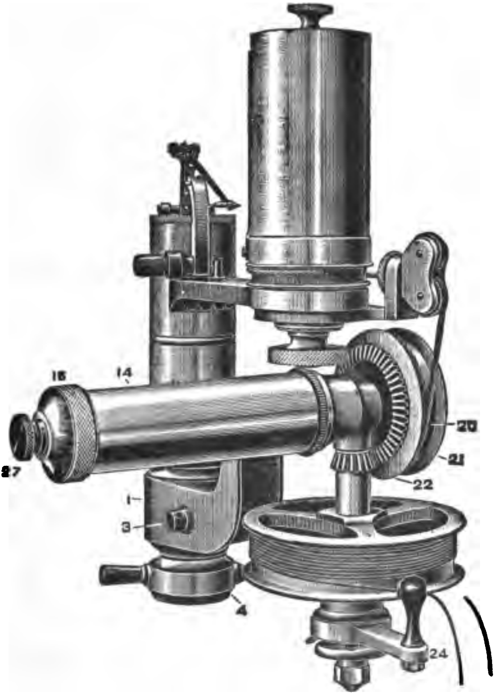


FIG. 16.

CROSBY REDUCING WHEEL.

The Crosby reducing wheel is attached directly to the cylinder cock of the steam engine, and has connected to it the steam engine indicator which it is to serve; thus it forms a base or support for the latter, and receives all the strains and shocks in the operations of the engine, to the relief of the indicator. All its parts are designed and constructed for strength, accuracy, and durability. Its bearings are not only nicely adjusted, but are made comparatively frictionless by the introduction of minute balls running in hardened tool steel bearings, affording lightness and freedom of movement. It has a helical spring which is more active in its purpose than the volute spring in common use, this being a very essential feature for accurate results on high speed engines. The cord pulley is horizontal to allow the cord leading to the engine cross-head to take any direction the circumstances may require without regard to the position of the indicator.

Patented.



This Cut Shows the Crosby Indicator Mounted upon the Crosby Reducing Wheel.

Whenever the reducing wheel is to be attached to a vertical engine an elbow nipple is provided, which will allow the cord pulley to travel in the proper plane for guiding it to the cross-head of the engine, with the indicator in an upright position as usual.

Special tools have been provided for making it, so that like parts are interchangeable, and when worn or destroyed others can be easily substituted; in other words, everything has been done so far as possible to make the instrument in all respects as excellent for its purpose as has been done in regard to the Crosby Indicator.

It is adapted to receive any steam engine indicator or indicator cock by means of interchangeable bushings; and by a series of speed pulleys to all steam engines having strokes between the limits of fourteen inches and seventy-two inches.

Directions for Operating the Crosby Reducing Wheel.

Attachment. Attach it directly to the cylinder cock of the steam engine by means of the union 4 of the standard 1. Connect the indicator to the standard 1 with the paper drum standing over the spring tube 14, and the indicator guide pulley in a proper position over the stroke pulley 20.

To Attach the Cord Guide. Loosen the cord guide 24 by means of the screw beneath the cord pulley and move it around to the proper position for the cord to pass directly through the hole in the cord guide without rubbing, to the crosshead of the engine; then tighten it in place.

To Take Up the Tension Spring. Release the thumb-screw 27 in the end of the shaft within the spring tube 14; withdraw the knurled spring head 16 from its square end, and turn it one or more squares as may be desired.

To Adjust the Stroke Pulley. Remove the knurled disc 21 holding the stroke pulleys 20 in place on the gear shaft; place thereon the stroke pulley desired; replace the disc, and screw it up firmly with the fingers.

To Attach the Indicator Cord. Adjust the indicator cord one or more turns around the stroke pulley 20, passing the end through the hole in and around the hook of the knurled disc 21.

When Used with Other Indicators. Loosen the bolt 3 in the side of the standard 1 where it is attached to the cylinder cock of the steam engine; remove the bushing and insert another fitted to the indicator to be used.

Important Suggestions.

In all cases the indicator cord should be of the right length to prevent the paper drum from recoiling against its stop; and before attaching it to the cross-head of the engine it should be drawn out its full length to ascertain whether or not the cords on the indicator and reducing wheel have been properly adjusted.

All the working parts must be kept well oiled.

The reducing wheel is adapted to be used with a steam engine indicator having either a 1½″ or 2″ drum. As indicators with 2″ drums are ordinarily used, the reducing wheel is provided only with stroke pulleys for such size drum. If the reducing wheel is to be used with an indicator having the 1½″ drum, it should be so stated in order to receive the stroke pulleys of the proper size.

These stroke pulleys are provided in sets and can be so obtained.

Bushings for attaching other than the Crosby indicators, and elbow nipples for attaching the reducing wheel to a vertical engine, are extra.

Whatever drum motion mechanism is used, its accuracy can be easily tested in the following manner: Lay off on the guides, points at, ¼, ½, and ¾ of the stroke. Connect the indicator with the drum motion in the same manner as for taking diagrams. When the cross-head is on either dead center, touch the pencil to the paper and make a vertical mark, and in the same way make vertical marks when the cross-head reaches each successive quarter point on the guides. If the marks are exactly at fourths on the card, the motion of the cross-head has been accurately reduced.

The directions here given for constructing and arranging drum motions are general; special cases may require modification of the forms and special adaptation of the means here described, all of which call forth the ingenuity and skill of the engineer.

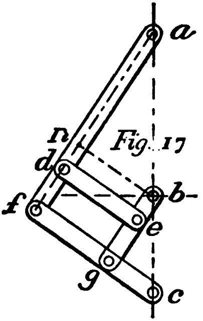


Fig. 17

Fig. *17* shows a pantograph device at mid-stroke. This is made of bar iron nicely riveted together. The indicator cord may be attached at *b*. The end *a* is attached to a pin on the cross-head. The fixed fulcrum is at *c*. *a*, *b*, and *c* must always lie in the same straight line, and *e d*, *b n*, parallel and equal to *f g*. Also, *a f* : *n f* = stroke of piston to length of indicator diagram.

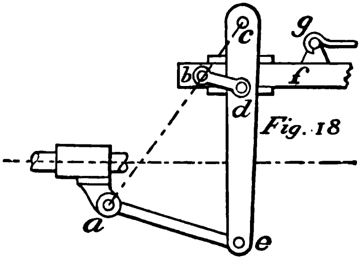


Fig. 18

Fig. *18* illustrates a device used at the Massachusetts Institute of Technology. *f* is a rod moving in a slide parallel to the piston-rod. The link *b d* is attached to *f*, and the link *a e* to the cross-head. *a*, *b*, and *c* must always lie in the same

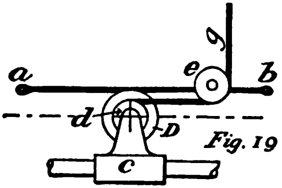


Fig. 19

straight line. *a e* : *b d* and *e c* : *c d* = stroke of piston to length of indicator diagram. The cord is hooked on a pin at *g*; it is well to have a pin for each indicator used.

Fig. 19 is a device by Armand Stévart for long strokes. *a* and *b* are fixed ends of cord wrapped around a pulley *D*. The indicator cord is attached to a small pulley *d* and passes around a guide pulley *e*. *D* and *d* are attached to the cross-head. Dia. *D* ÷ dia. *d* = stroke piston ÷ by the difference between stroke of piston and length of card.

CHAPTER VIII.

HOW TO TAKE DIAGRAMS.

First connect the indicator to the indicator cock.

Adjust the guide wheels under the drum so that the cord leads from this wheel in the right plane.

All reducing motions of the pantograph type, such as the lazy tongs shown by Fig. 16 and the modifications shown by Figs. 17 and 18, require for correct reduction of motion that the string to the indicator should run in a line parallel to the line of motion of the piston-rod.

With reducing motions of the Brumbo pulley type, shown by Fig. 15, it makes no difference at what angle the string leads from the guide wheel to the arc. The string must, of course, be in the plane of the arc.

After the guide wheels under the drum have been adjusted and fastened, next change the location of the hook on the drum string so that when the hook is pulled way out as far as it will go, it overlaps the pin or ring or loop or whatever is provided for a connection on the reducing motion, by about 3/8 of an inch. Then let the drum spring pull the hook way back and note how near the hook comes to the pin or loop on the reducing motion. If the hook is drawn back 3/8 of an inch beyond the travel of the pin or loop on the reducing motion the cord is the right length. Should the distance at this end be 1 inch and at the other 3/8 of an inch the cord should be lengthened so as to make the distances at the two extremes alike.

The paper is now put on the drum. The paper must be tight and pushed down to the bottom of the clips.

There are two ways in which the paper may be held by the clips.

The ends of the paper may be brought out in the center between the two clips, or the two clips may be used as one piece and the two ends lapped under the clip. Next connect the drum to the reducing motion.

If one understands how to do this, it makes no difference whether the engine is making 10 or 600 revolutions per minute.

Oftentimes it is amusing to watch one who does not know how, try to catch the loop in the string from a Brumbo pulley reducing motion on a high speed engine.

With one hand pull the hook on the indicator cord out as far as it will go. With the other hand take hold of the string from the Brumbo pulley and let the string be pulled through your fingers till the loop is reached. Hold the loop so that each time the engine reaches the crank end of its stroke you will feel a slight pull on the loop due to the winding of the string on the arc.

The hook on the indicator cord will reach ⅜ of an inch beyond the end of the loop, making it easy to connect. The speed of the engine, it will be seen, does not make any difficulty about connecting on if this method is used.

Steam is now turned on the indicator through the indicator cock, and after a period of ten seconds, in which time the indicator is being heated up by the steam, a card may be taken by pressing the wooden handle moving the pencil mechanism up against its stop. After taking the card, close the cock and draw an atmospheric line. Then disconnect the drum string from the reducing motion.

Should the lines on the card be too faint the wooden handle may be screwed back a turn or two.

Formerly, graphite was used for the marking point and ordinary paper as drum paper.

To-day metallic paper (a paper coated with a salt of lead) is almost universally used and the marking point is a piece of soft brass.

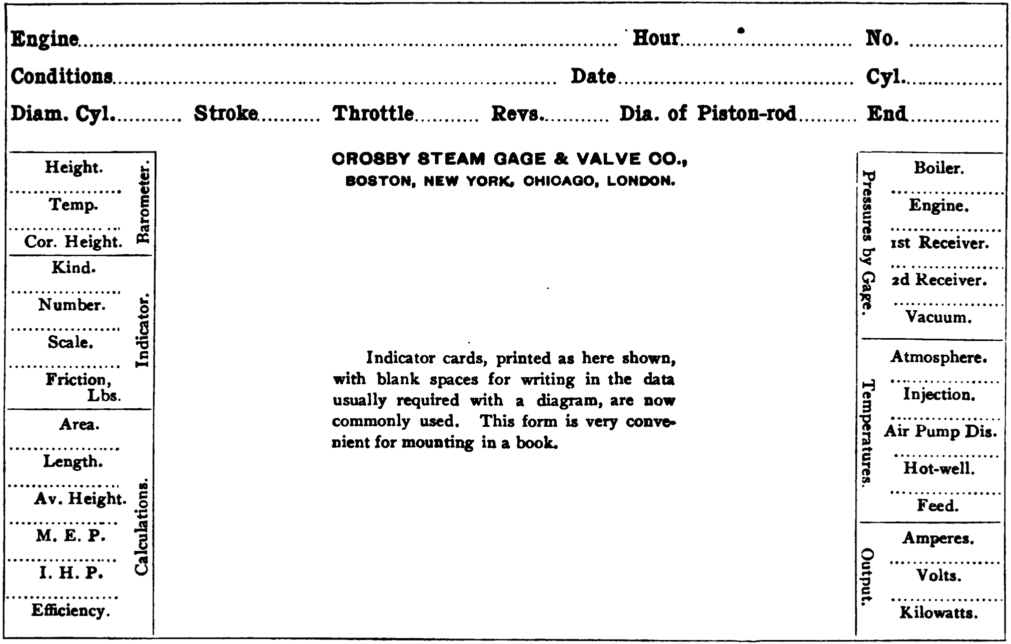
This makes it possible to get very fine lines on the card and to take cards with very little pressure on the marking point. Any friction between the pencil and the paper makes an error in the card, and cards should be taken with as faint lines as possible on this account.

Make notes on the card of as many of the following facts as possible. The day and hour of taking the diagram; the kind of engine from which the diagram is taken, which end of the cylinder and which engine, if one of a pair; the diameter of the cylinder, the length of the stroke, the diameter of the piston-rod, the number of revolutions per minute, and the position of the throttle. The atmospheric pressure; the steam pressure at the boiler and at the engine, by the gage; the vacuum by the gage on condenser and the temperature of the feed at the boiler; if the engine is compound, the pressure in the receiver; the scale of the spring used in the indicator; the volume of the clearance at each end of the cylinder, and what per cent. of the piston displacement each of these volumes is. (Directions for ascertaining the *volume* of the *clearance*, and what *per cent.* of that volume is of the *piston displacement*, are given on pp. 91, 92, and 93.)

It is often useful to make notes of *special* circumstances of importance, such as a description of the boiler, the diameter and length of the steam and exhaust pipes, the temperature of the feed water, the quantity of water consumed per hour, etc.

On a locomotive, note the time of passage between mile-posts in minutes and seconds, from which, when the diameter of the drivers is known, the number of revolutions per minute may be calculated. Note also the position of the throttle and the link, the size of the blast orifice, the weight of the train, and the gradient.

On diagrams from marine engines note, in addition to the general facts, the speed of the ship in knots per hour, the direction and force of the wind, the direction and state of the sea, the diameter and pitch of the screw, the kind of coal, the amount consumed, and the ashes made per hour.



Engine.................... **Hour**.......... **No.** ....
Conditions.................... **Date**.......... **Cyl.**..........
Diam. Cyl........... **Stroke**.......... **Throttle**.......... **Revs.**.......... **Dia. of Piston-rod**.......... **End**..........

Barometer.
Height.
Temp.
Cor. Height.

Indicator.
Kind.
Number.
Scale.
Friction, Lbs.

Calculations.
Area.
Length.
Av. Height.
M. E. P.
I. H. P.
Efficiency.

CROSBY STEAM GAGE & VALVE CO.,
BOSTON, NEW YORK, CHICAGO, LONDON.

Indicator cards, printed as here shown, with blank spaces for writing in the data usually required with a diagram, are now commonly used. This form is very convenient for mounting in a book.

Pressures by Gage.
Boiler.
Engine.
1st Receiver.
2d Receiver.
Vacuum.

Temperatures.
Atmosphere.
Injection.
Air Pump Dis.
Hot-well.
Feed.

Output.
Amperes.
Volts.
Kilowatts.

CHAPTER IX.

HOW TO FIND THE POWER OF AN ENGINE.

To find the power actually exerted within the cylinder of a steam engine, it is necessary to ascertain separately *three factors* and the product of their continued multiplication. These factors are: The area of the two sides of the piston; the total travel of the piston in feet per minute; and the mean effective pressure urging the piston forward, designated by M. E. P.

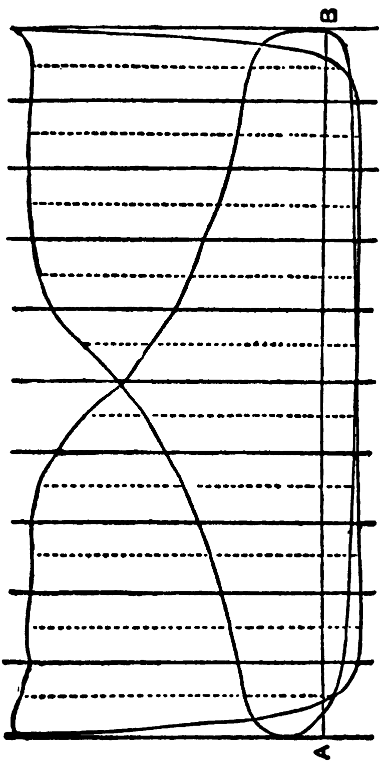
The Piston Area. This, at the back end, is the same as the area of cross-section of the cylinder; at the crank end it is the same, less the area of cross-section of the piston-rod. These areas may be found from their diameters in a table of the areas of circles, or be computed by multiplying the square of the diameter in inches by the approximate number 0.7854.

The Travel of the Piston. The total travel of the piston in feet per minute is found by multiplying twice the length of the stroke measured in *feet*, by the number of revolutions of the crank shaft per minute, which should be carefully ascertained by taking the mean of many countings, or the readings of a speed counter during a considerable time. The mean piston speed will be expressed in terms of *feet per minute.*

The Mean Effective Pressure. There are several approximate methods for computing the mean effective pressure, one of which is to divide the diagram into ten equal parts, as shown in Fig. 20. Then through the points of division draw lines, which are called ordinates, at right angles to the atmospheric line. The mean heights or pressures of the small areas thus formed are indicated by the dotted lines midway between the ordinates.

The mean effective pressure of the whole (of each) diagram may now be found, by measuring (on the dotted lines) the mean pressure in each of the small areas with the scale corresponding to the spring used in taking the diagram.

FIG. 20.



Diagrams from Hartford Engine. Cylinder, 16 x 24 inches. Boiler pressure, 87 lbs. Vacuum per gage, 23½ inches. 130 revolutions per minute. Scale, 50.

The sum of these mean pressures, divided by 10, the number of divisions, will give the mean effective pressure sought, in lbs. per square inch.

If a diagram has many irregularities of outline, it may be necessary to divide it into twenty equal divisions to insure a correct measurement of the pressures; in such a case we divide the sum of the pressures by 20 instead of 10. In other cases, when irregularities occur only in a part of a diagram, it is only necessary to subdivide one or more of the ten divisions to insure greater accuracy in that part; in such a case we must measure the pressure in each subdivision and divide their sum by 2 to get the mean pressure of that division. (See Fig. 22 for a full illustration of this method.)

If the scale is not at hand, the heights of the divisions may be pricked or marked off on a strip of paper, one after the other continuously until all are measured; then the distance from the end of the strip to the last mark will represent the sum of all the measurements, which can be measured in inches with an ordinary rule. This quantity, divided by the number of divisions in the diagram — or diagrams, if there are two — and multiplied by the scale of the spring used, will give the average or mean effective pressure, the same as by the other method.

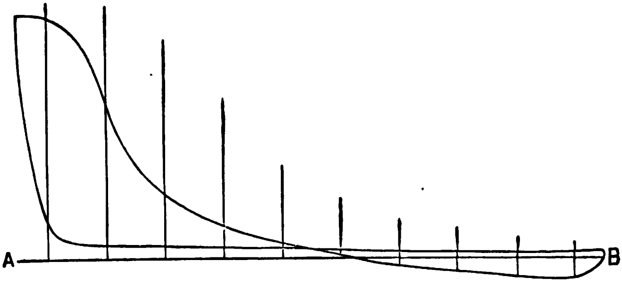


Fig. 21.

When there is a loop in the diagram as in Fig. 21, the area enclosed in the loop should be subtracted from the other part, as it represents loss of efficiency.

The quickest and most accurate method for measuring the diagram and finding the mean effective pressure is by the use of Amsler's Polar Planimeter.

With careful manipulation, the planimeter will give the *exact area* of a diagram in square inches and decimal parts thereof, to hundredths of a square inch, and the tedious process of dividing the diagram into equal parts and measuring their average pressures or heights, with the liability of making errors, is avoided.

Measure the diagram with the planimeter as directed in Chapter IV. Divide the number of square inches area thus found by the length of the diagram, expressed in inches and decimals, and the result will be the average height of the diagram. Multiply this average height by the scale corresponding to the spring used in taking the diagram and the result will be the mean effective pressure. It is better to multiply first and divide afterwards to avoid troublesome fractions. (A description of the planimeter and full directions for its use on indicator diagrams are given in Chapter IV.)

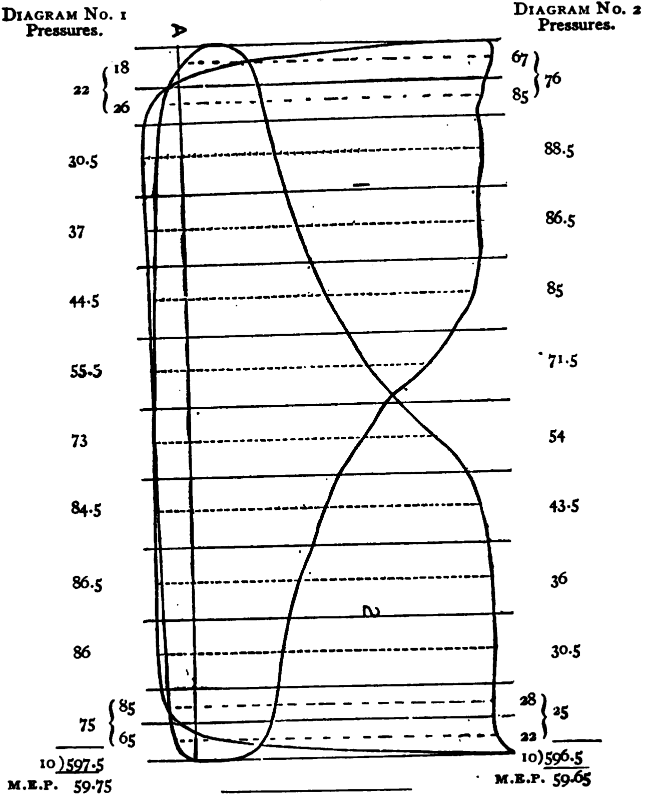
Fig. 22 illustrates two diagrams divided first into ten equal spaces and then *each end space subdivided* so as to more accurately measure those parts of each in which the greatest irregularities occur. Observe that the pressures or heights of the subdivisions of each end space are measured, and the sum of these measurements divided by two to get the mean pressure or height of that one of the ten spaces.

The pressures of Diagram No. 1, as measured by the scale, are set in a column on the left, while those of No. 2 are set in a column on the right. The sum of each column divided by ten gives the M. E. P. of that diagram.

The heights of Diagram No. 1 marked off on a slip of paper continuously, measure 11.93 inches, while those of No. 2 measure 11.95 inches; these quantities, divided by 10 and multiplied by 50, give the M. E. P. of each diagram respectively, and if accurately measured, will be the same as found by the scale.

These diagrams, when measured by the planimeter, give results which are substantially the same

FIG. 22.



HEIGHTS OF DIVISIONS MEASURED ON A STRIP OF PAPER.

DIAGRAM NO. 1.		DIAGRAM NO. 2
10)11.95 in.	Divide by 10	10)11.93 in.
1.195		1.193
50	Multiply by proper scale.	50
M. E. P. 59.750		M. E. P. 59.650

PLANIMETER MEASUREMENTS.

	DIAGRAM NO. 1.	DIAGRAM NO. 2.
Square inches,	4.42	4.46
Length,	3.72	3.73
Average height,	1.188	1.195
M. E. P.,	59.4 lbs.	59.75 lbs

as found by the approximate methods. These results are given at the bottom of page 84.

CALCULATING I. H. P.

Let a = the area of the head end of the piston in square inches.

Let r = the area of the piston-rod in square inches.

Let s = the stroke of the engine in feet.

Let n = the number of revolutions per minute.

The I. H. P. or indicated horse-power of the head end of the cylinder is

$$\frac{a \times (\text{M. E. P. of head end}) \times s \times n}{33{,}000}$$

The I. H. P. of the crank end is

$$\frac{(a - r) \times (\text{M. E. P. of crank end}) \times s \times n}{33{,}000}$$

The I. H. P. of the cylinder is the sum of these two.

If the M. E. P. and the revolutions per minute are each made equal to 1 in the preceding expressions, then the result obtained from each is the I. H. P. per pound M. E. P. at one revolution per minute. These factors are called the *engine constants*. The *engine constants* for the two ends are respectively

$$\frac{a \times s}{33{,}000} \text{ and } \frac{(a - r) \times s}{33{,}000}$$

If the valves of an engine are well adjusted and the M. E. P. for the two ends of the cylinder is nearly the same, an approximate calculation of the indicated horse-power of the cylinder may be made by multiplying the average M. E. P. of the two ends by the number of revolutions per minute and by the sum of the engine constants.

CALCULATING HORSE-POWER FROM GAS ENGINE CARDS.

Otto Cycle.

A gas engine working on the Otto Cycle may, according to the method of governing, have explosions coming repeatedly every fourth stroke, or the explosions, which cannot in any case come oftener than the fourth stroke, may be skipped for one, two, or three periods of four strokes.

No explosive mixture is taken into the cylinder at such times as the explosions are skipped.

Where the explosion comes regularly every fourth stroke the governing is commonly done by varying the amount of explosive mixture taken into the cylinder, and by this means the area of the card and the mean effective pressure.

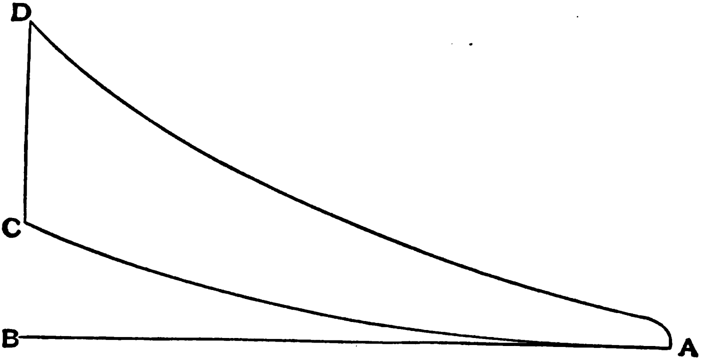


FIG. 23.

In calculating the power of a gas engine the mean effective pressure is figured as already explained.

The M. E. P. times the area of the piston in square inches, gives the total mean pressure on the piston. This is now to be multiplied by *the travel of the piston in feet per minute during working strokes* and the result divided by 33,000.

In the case of an engine governing by the "hit-and-miss" method some form of ratchet counter should be attached to the gas inlet valve so that the number of explosions per minute can be obtained.

Example: The card shown by Fig. 23 has a mean effective pressure of 85 pounds. Diameter cylinder 11″; stroke 24″; revolutions per minute, 230; explosions per minute, 35.

The total travel of the piston in feet per minute during working strokes is 35 × 2 = 70 feet. The area of the piston 11″ in diameter is 95.03 square inches.

$$\frac{95.03 \times 85 \times 70}{33{,}000} = 17.1 \text{ H. P.}$$

A gas engine governing by the "hit-and-miss" method may have its firing cycle run along as follows:—

Light load—fire 1, miss 5; fire 1, miss 5, etc.
Medium load, fire 3, miss 2; fire 3, miss 2, etc.
fire 2, miss 2; fire 3, miss 2, etc.
fire 3, miss 3; fire 1, miss 1, etc.
Full load—fire 7, miss 2; fire 7, etc.

If the engine is running with a steady load the firing cycle may repeat itself for a period of an hour or longer, then it may change to some other cycle.

The pencil of the indicator should be left on the paper long enough to get the cards for a complete firing cycle.

This may mean that there will be seven cards on the same paper. It will be noticed, however, that the cards are of different size.

The card taken after a miss will be larger than one taken in the center of a period of three fires.

The card taken after a number of misses is always larger than the other cards. This is explained by the fact that the clearance space of a gas engine, amounting to about 35 per cent. of the piston displacement, is left partially filled with carbonic acid gas at the end of the exhaust stroke.

If a new firing charge of gas and air is at once taken in, it becomes mixed with this large amount of carbonic acid gas, and its explosive force is dampened. If, however, there have been two or three misses preceding this intaking of the explosive mixture, the air (which is always taken at every fourth stroke) will have diluted and cleared out much of this carbonic acid gas, so that nearly the full force of the explosion will be obtained.

To calculate the power from the indicator cards, the mean area of all the cards in a firing cycle should be taken.

Oftentimes this mean area may be obtained by planimetering that card which appears to represent an average of the whole number in the cycle.

The spring used in a gas engine indicator is seldom less than a 200-pound spring, and often of a higher rating. With a spring as stiff as this it is not possible to detect differences in pressure not greater than 2 or 3 pounds.

To find the resistances through the intake and exhaust values, a light spring (20 pounds) may be used in the indicator. To prevent the spring from being injured by the pressure developed at explosion, a stop in the shape of a thin brass ferrule should be put outside of the spring and between the piston and the cap. After the spring has been compressed a certain amount the piston brings up on the stop, thus preventing any more load from coming to the spring.

Referring to Fig. 23 it will be seen that the line A B, the exhaust line, and the line B A, the filling or intake line, and the atmospheric line, all appear as one line when a stiff spring is used.

The light spring will show the line A B slightly above the atmospheric line, and the line B A slightly below it.

CALCULATING HORSE-POWER FROM GASOLINE OR GAS ENGINE CARDS.

Two Cycle.

The two cycle engines have a working stroke on the stroke the piston makes towards the crank.

The burned gases are driven out at the end of this stroke partly by their own pressure and partly by the inrush of a fresh charge which has been slightly compressed in the closed crank case during the working stroke of the piston.

The engine has one working stroke per revolution.

A certain amount of power is used up in drawing in and in slightly compressing the mixture in the crank case.

For accurate work an indicator with a 10 spring, or, preferably a lighter spring, should be attached to the crank case and the power so used deducted from that shown by the working stroke.

DISCUSSION OF M. E. P.

The indicator card from one end of a cylinder shows the pressures on that end *during a revolution*, or a stroke forward and a stroke back.

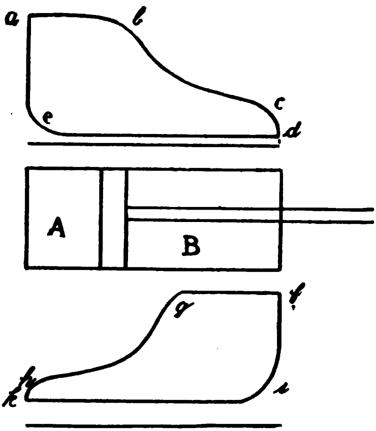
The Mean Effective Pressure, M. E. P. (improperly named perhaps) used in figuring horse-power is calculated from the indicator cards as already explained.

If the average push or pull given the piston-rod be divided by the area of the piston the result is not the same as the M. E. P.; correction being made for the difference of areas, if any, of the two sides of the piston.

(See discussion of stroke cards given later on.)

That the horse-power obtained is correct when the M. E. P. is calculated as previously explained is shown by the illustration on the next page.

Let the mean pressure above the atmospheric line corresponding to line $a\ b\ c\ d = P_f$.



Let the mean pressure above atmospheric line corresponding to line $f\ q\ h\ k = P_b$.

Let the mean pressure above the atmospheric line corresponding to $k\ i\ f = E_b$.

Let the mean pressure above the atmospheric line corresponding to $d\ e\ a = E_f$.

Let $A =$ area head end of piston in square inches.

Let $B =$ area crank end of piston in square inches.

Let $S =$ stroke in feet.

The total mean pressure on the piston-rod during the forward stroke of the piston is

$$A\ P_f - B\ E_b.$$

The total mean pull on the piston-rod during the return stroke is

$$B\ P_b - A\ E_f.$$

The total foot pounds per revolution is

$$A\ S\ P_f - B\ S\ E_b + B\ S\ P_b - A\ S\ E_f$$

$$\text{or } A\ (P_f - E_f)\ S + B\ (P_b - E_b)\ S.$$

$P_f - E_f$ is the mean effective pressure of the forward card.

$P_b - E_b$ is the mean effective pressure of the crank card.

CHAPTER X.

THE HYPERBOLIC CURVE.

This curve is frequently applied to indicator diagrams for the purpose of comparing it with the expansion curve as drawn by the indicator, and if it coincides pretty nearly, this fact may generally be taken as evidence tending to show that the steam and exhaust valves of the engine are properly closed and the piston tight.

Without going into any discussion regarding condensation and re-evaporation in steam engine cylinders, it is a well-known fact that indicator diagrams, taken from large engines, properly made and in good order, show expansion curves which are close approximations to the hyperbola.

Before this curve can be drawn, it is necessary to ascertain the capacity of the clearance or waste room: that is, all the space between the cylinder heads and the piston at each dead-center, including the counterbore and the ports up to the face of the closed valves.

There are several ways of finding this: — One, by direct calculation from sectional drawings, when accurate drawings can be obtained; another, by putting the engine at dead-center with valves closed, and then filling the clearance space with water, which has been carefully weighed in a convenient vessel, then weighing what is left. The difference between the weight of the whole and the remainder is the weight of water required to fill the clearance space. From this the number of cubic inches occupied by the water may be computed. At ordinary temperatures (60° to 75° F.), for all practical purposes, we may call the weight of one cubic inch of water 0.036 lbs. and 27.8 cubic inches of water equal to one pound. Then the number of pounds of water divided by 0.036 or multiplied by 27.8 will give the number of cubic inches. If accurate scales for weighing the water are not at hand, it can be carefully measured in a quart or pint measure and

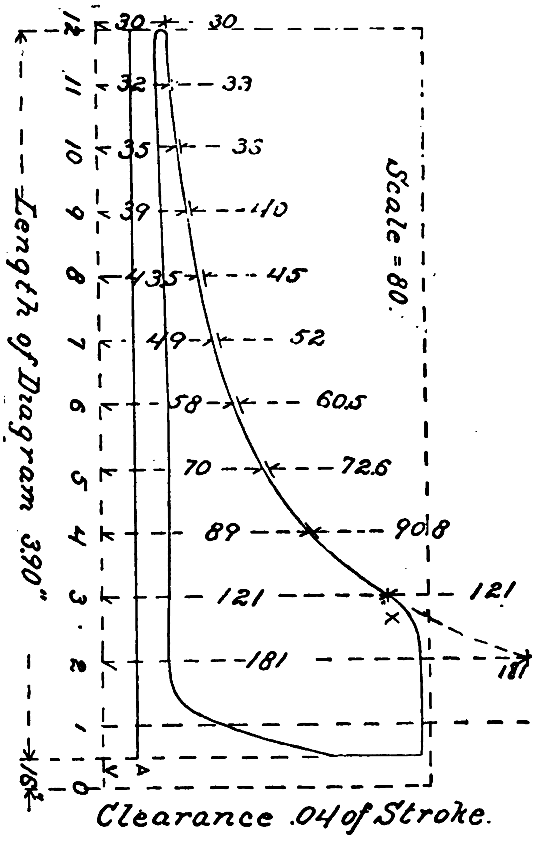


FIG. 24.

the number of cubic inches found directly. A gallon contains 231 cubic inches, a quart 57.75, and a pint 28.875 cubic inches.

The volume of the clearance will rarely be alike at the two ends of the cylinder, therefore the number of cubic inches in the clearance at each end must be divided by the net area of the piston at its own end: that is, the number of cubic inches in the clearance at the end nearest the crank must be divided by the number of square inches in the cross-section of the cylinder, less the number of square inches in the cross-section of the piston-rod; and the number of cubic inches in the clearance at the end farthest from the crank must be divided by the number of square inches in the cross-section of the cylinder. The quotient in each case will be the length of clearance at the respective ends of the cylinder, expressed in inches.

It is convenient to have the length of the clearance expressed as a fraction of the piston displacement or stroke of the piston. To obtain this fraction, divide the number of cubic inches in volume of clearance by the number of cubic inches in the volume swept through by the piston at each end separately, taking care to allow for the volume occupied at one end by the piston-rod, and the quotient will be the decimal fraction that the clearance space is of the volume swept through by the piston. In this instance (Fig. 24) it is found to be .16 inches.

Fig. 24 illustrates a good method for locating points in the hyperbola through which the curve may be drawn.

First, draw the zero line V, at the proper distance, viz. : $14\frac{7}{10}$ lbs. by the scale below and parallel with the atmospheric line ; next, draw the clearance line O, as computed, .16 of an inch from the end of the diagram ; next, locate the point of cut-off X, and draw the perpendicular line numbered 3 through it ; next, divide the space between this line and the clearance line into three equal parts ; then, taking one of these parts for a measure, point off, on the vacuum line, *equal* spaces toward the left hand until one or more falls beyond the end of the diagram as shown, and erect perpendicular lines from each point. These lines are called ordinates and numbered consecutively 1, 2, 3, 4, etc., beginning with the one next to the clearance line. It is well to bear in mind the fact that vertical distance on a diagram represents *pressure* and horizontal distance *volume*.

In this case we have started the hyperbola from the point of cut-off X, and its course is indicated by the short lines drawn through the ordinates a little above the actual curve, with their calculated pressures written above; the actual pressures of the expansion curve are written below it. The properties of the hyperbola are such, that if the distance of the point X from the clearance line O be multiplied by the height of X from the zero line V, the height of any other point in the curve can be found by dividing this product by its distance from the clearance line. And on this principle we proceed to locate points on the ordinates through which our hyperbola will run.

We find the pressure at the point of cut-off to be 121 lbs. with a volume which we call 3, because there are three spaces or volumes between it and the clearance line. Then, 121 × 3 = 363, which is our dividend for all the other volumes. Therefore, the height at which the hyperbola will cut ordinate 4, will be determined by dividing 363 by 4, which is 90.8 (it is unnecessary to carry the division beyond one decimal) and of ordinate 5, 72.6; of ordinate 6, 60.5; and so on to the end. At ordinate 12 we find that the hyperbolic and the actual curves practically coincide. In like manner we may extend the curve to the right: 363 ÷ 2 = 181 lbs., which would be the pressure if the steam were compressed up to 2 volumes. If desired, the hyperbolic curve can be started just before the point of release and projected in the opposite direction by the same method.

Instead of using figures which stand for pressures or volumes of steam to locate the hyperbola as in this instance, the distances from the base and perpendicular lines of any point may be expressed in inches and decimal parts, with the same result.

A quick way to draw the hyperbola is to take the whole distance between ordinate 3 and the clearance line as a measure, and set off equal spaces to the left as before directed. Then we would have but four ordinates and would number them as follows: 1 at 3d, 2 at 6th, 3 at 9th, and 4 at 12th. At 1 we would have a pressure of 121 lbs.; at 2, 121 lbs. $\div 2 = 60.5$; at 3, 121 lbs. $\div 3 = 40$; and at 4, 121 lbs. $\div 4 = 30$.

As a general rule, the near approximation of the expansion curve to the theoretical or hyperbolic curve may be taken as evidence of good conditions but should not be accepted for a certainty, unless all the known facts and conditions tend in the same direction.

GEOMETRIC METHOD OF FINDING THE HYPERBOLA.

The hyperbola may be found by following the directions given below, in connection with Fig. 25. A is the atmospheric line; Z the zero line, or line of no pressure; B the line of boiler pressure, and C the clearance line. Locate the first point in the hyperbola at the point of release X and draw the vertical line X E. Then draw diagonal line, E H; then, from X, draw horizontal line 5 to its intersection with E H, through which draw vertical line F O. Now mark off points between O and E as 1, 2, 3, 4, — exact spacing is unnecessary — and from these points draw diagonal lines to H and vertical lines down to, or a little below, the actual curve. Now draw horizontal lines 6, 7, 8, and 9 from the points of intersection in the line F O, of the diagonal lines H 4, H 3, H 2, and H 1 respectively; and the points where these lines cross the vertical lines, 1, 2, 3, and 4, in connection with points X and O, are the points through which the hyperbola should be drawn, as shown by the dotted curve.

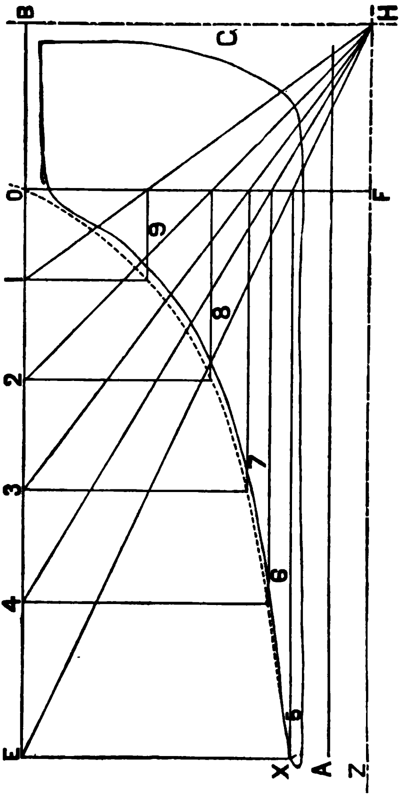


Fig. 25.

ANOTHER METHOD OF FINDING THE HYPERBOLA.

(See Fig. 26.)

Through the point of release *b*, draw any line as *a* B and make A B equal to *a b*. Then draw any other line as *c* D and make *c d* equal to A D; then *d* will be a point in the hyperbola passing from *b* to A, as shown by the dotted curve. By drawing a number of lines through A and following the same method, we can find as many points in the hyperbola.

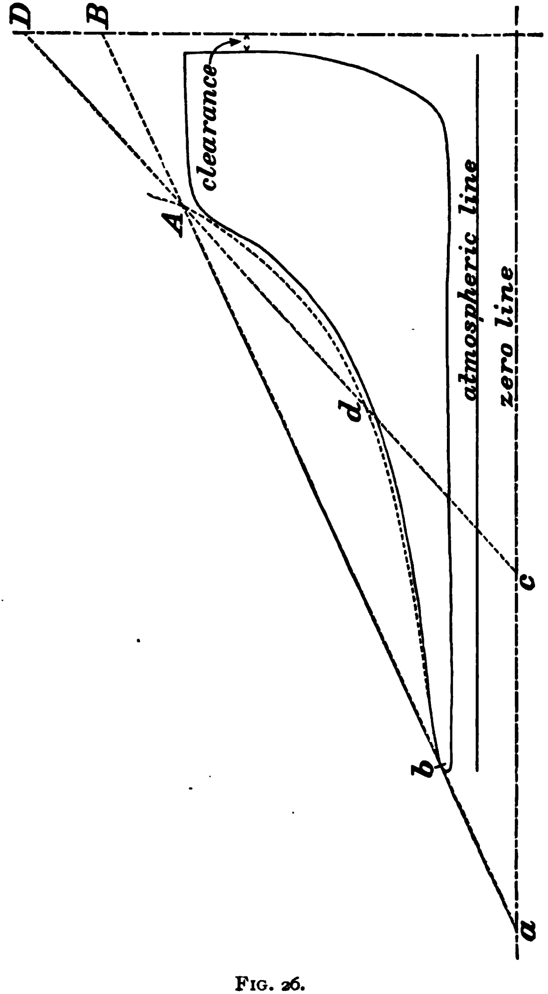


FIG. 26.

PART II.

PART II. APPLICATIONS.

The steam engine indicator may be used for a great many purposes besides those mentioned in the preceding chapters. The applications are so varied that the cases selected and illustrated in these pages show only a few of those which seem to be most useful.

1. Valve Setting.

If the cards taken from the two ends of the cylinder resemble Figs. 1 or 2, the eccentric has too little angular advance and must be turned ahead; in the case of an engine giving cards shown by Fig. 1, moving the eccentric ahead 5° to 10° would probably be sufficient, while with Fig. 2, 20° to 30° would be needed.

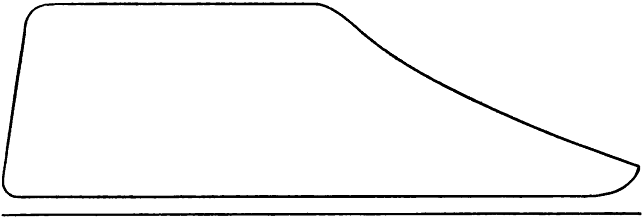


Fig.-1.

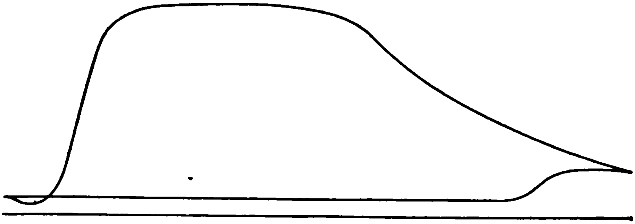


Fig.-2.

If the cards from both ends resemble Figs. 3 or 4, the eccentric has too much angular advance and should be turned back a small amount for Fig. 3, and a much larger amount for Fig. 4.

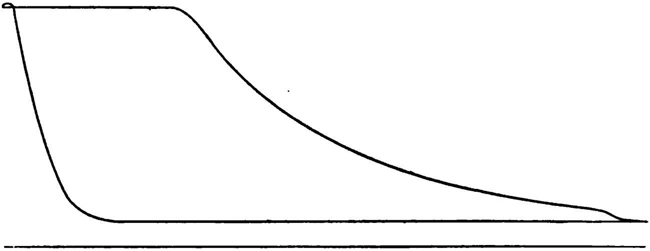


Fig.-3.

Cards similar to Figs. 3 and 4 are obtained at early cut-offs from most single valve high speed engines with fly-wheel governors.

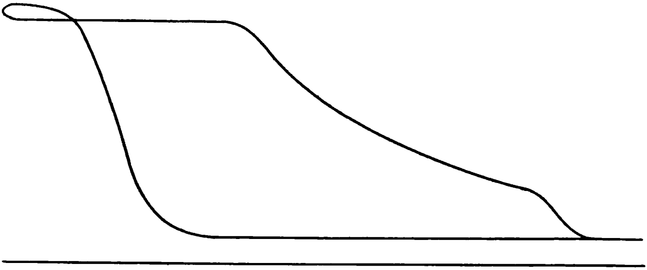


Fig.-4.

A plain slide valve engine, or an engine with piston valve, taking steam on the outside, will give cards like Figs. 5 and 6 when the valve spindle is too long, the eccentric being in the right place.

The effects of lengthening the valve spindle are (1) to increase the steam lap on the head end, delaying admission and hastening cut-off on this end; (2) to decrease the steam lap on the crank end, hastening admission and delaying cut-off; (3) to decrease the exhaust lap on the head end, hastening release and delaying compression; and (4) to increase the exhaust lap on the crank end, delaying release and hastening compression.

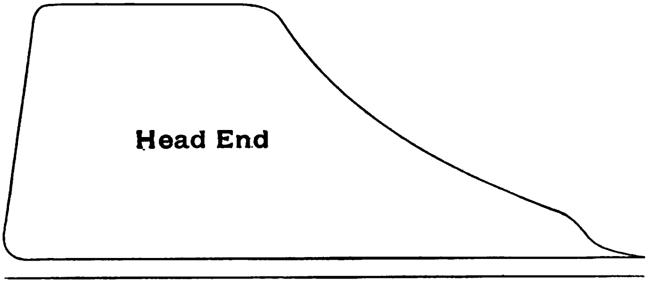


Fig.- 5.

If the valve spindle is too long and at the same time the eccentric has too little angular advance, or is set too far back, the cards from the engine will be similar to Figs. 7 and 8.

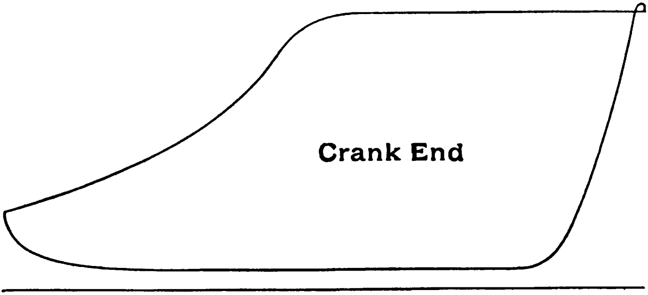


Fig.- 6

If the eccentric is set too far ahead and the spindle is too long, the appearance of the cards may be pre-

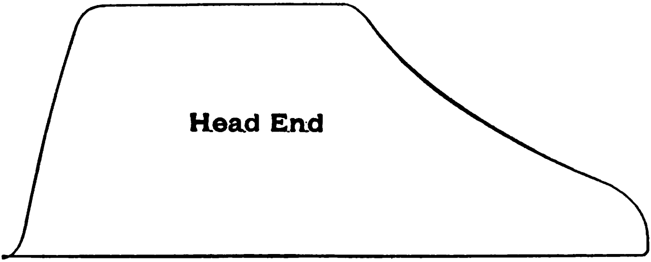


Fig.-7

dicted by combining the effects shown by Figs. 3 or 4 with Figs. 5 and 6.

2. Steam Chest Cards.

A steam engine indicator may be connected to the steam chest of the engine and cards taken simultaneously with cards from the cylinder, both indicators being attached to the same reducing motion.

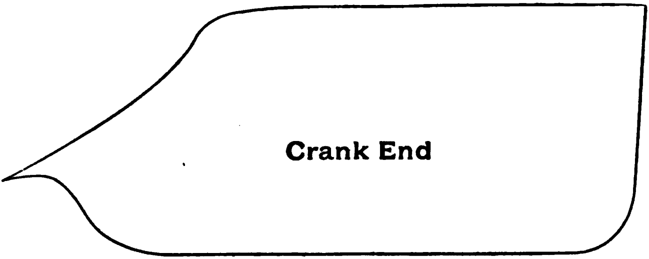


Fig.-8.

For comparison, it is convenient to superimpose the chest card on the cylinder card as has been done in Fig. 9.

The steam chest card A B C D E C A, Fig. 9 is for one revolution. If the steam pipe and steam port leading from the steam chest to the cylinder are large enough, there will be no appreciable drop in pressure between the boiler and the steam chest, or between the chest and the cylinder. If, however, a chest card similar to that shown by the full line is obtained, the steam pipe is too small. This is shown by the fact that the drop from boiler pressure A appears in the steam chest card while the drop between the steam chest and the cylinder is slight.

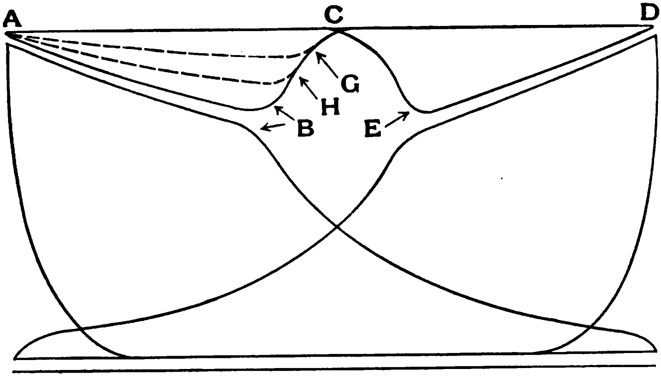


Fig.- 9.

If the chest card showed a slight drop in pressure, as indicated by the dotted line AG, it would mean that the steam port leading from the chest to the cylinder was too small.

A chest card shown by the line AH would mean that both the steam pipe and the steam ports were too small.

Sometimes the pressure at C is greater than boiler pressure, A. This results from the sudden checking of the velocity of the steam in the pipe supplying the engine.

3. The Eccentric Card.

Near the ends of the stroke of an engine the crank turns through a considerable number of degrees without giving much motion to the crosshead.

On account of this fact the indicator card taken in the ordinary way is of little value in investigating any peculiarities which may be noticed at or near the ends of the stroke.

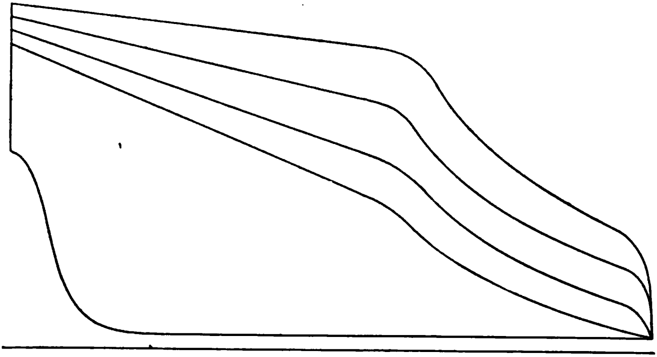


Fig.-10.

If, however, the drum motion be taken from the eccentric, which is ordinarily a little over 90° ahead of

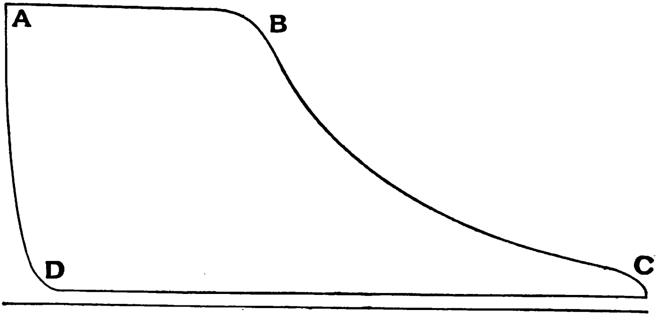


Fig.-11.

the crank, the compression and admission lines and the line at release will be spread out at the centre of the diagram, while the expansion and exhaust lines will be shortened and appear at the ends of the cards.

Fig. 11 represents a good steam card, and Fig. 12 an eccentric card.

The cards are lettered at admission, cut off, release, and compression with the letters A, B, C, D, respectively.

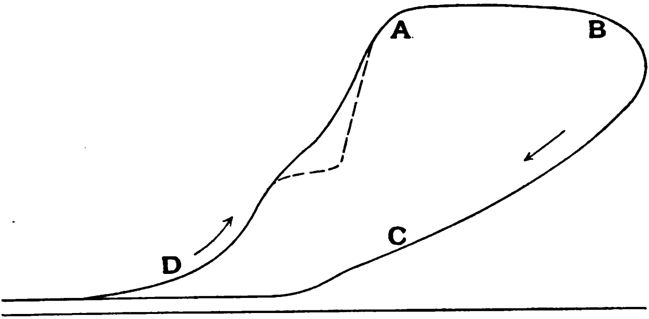


Fig.-12.

It will be noticed on the eccentric card that the line DA is a reverse curve.

On the card taken from an engine with throttling governor, Fig. 10, a peculiar drop near the end of compression is found. This is due partly to a leakage of steam from the clearance space over the bridge into the exhaust and partly to a slight amount of condensation. This drop appears on the eccentric card as in the dotted line.

4. Steam Cards from a Westinghouse Air Brake Pump, or from a Wet Air Pump with Surface Condenser.

In the diagram shown by Fig. 13 the piston starts at the left and moves towards the right. As the work opposed to the steam piston by the air piston is but trifling during the first part of the stroke, the steam piston "runs away from the steam," causing the drop in pressure shown.

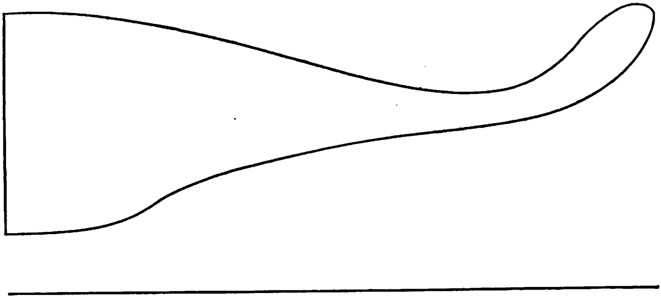


Fig.- 13.

Near the end of the stroke the extra work opposed to the steam piston by the air piston causes it to slow down, and then steam has time to fill the cylinder at nearly boiler pressure. On the return stroke the piston, during the first part of its stroke, travels so fast that the steam cannot be freely exhausted. During the latter part of the stroke, however, when the piston slows up, the pressure drops nearly to that in the exhaust pipe, as shown by the lower line of Fig. 13.

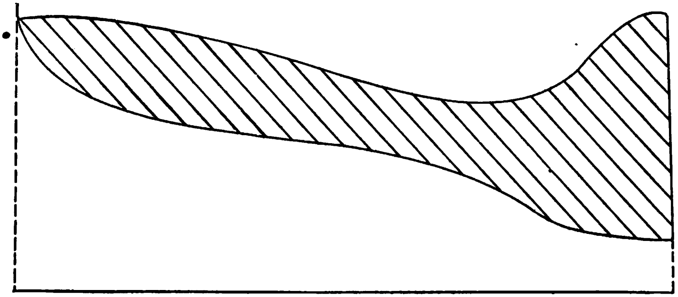


Fig.- 14.

A casual inspection of Fig. 13 might lead to the conclusion that the greatest effective pressure on the piston was at the left-hand end near the beginning of the stroke. This is not the case, as may be seen by constructing what is called the stroke card shown by Fig. 14, and again by Fig. 20. Assuming that the piston has a tail-rod so that the area of the head end of the piston is the same as the area of the crank end, the effective push per square inch on the piston at any time is the difference of pressure on the two sides at that time. If now the steam line of the card from the head end be combined with the back pressure line from the card of the other end, we get a diagram known as the Stroke Card, sometimes called the True Card, which, by its distance between lines, gives the effective pressure per square inch on the piston at any point. Such a diagram is shown by Fig. 14 for the Air Brake Card. It will be noted that the greatest effective push on the piston comes at the *right*-hand end, not the left.

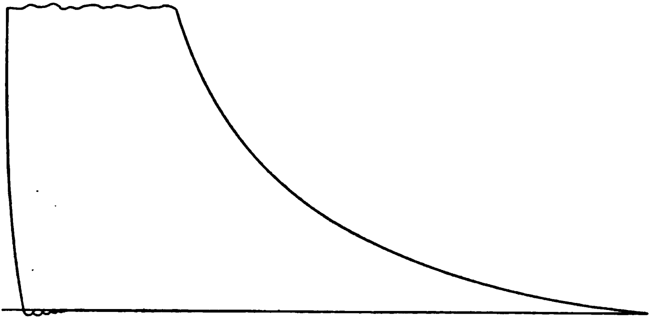


Fig.-15.

If the two ends of the steam piston have not the same area, the pressures from one indicator card may be multiplied by the ratio of the two areas, before plotting, in order to reduce to the same basis for comparison.

5. Air Compressor Cards.

Figs. 15 and 16 are from an Air Compressor and from the air end of an air pump, respectively.

The irregularities in the delivery line of Fig. 15 are due to the vibrations of the delivery valve. The slight drop at the beginning of air intake results from resistance to opening offered by the suction valves.

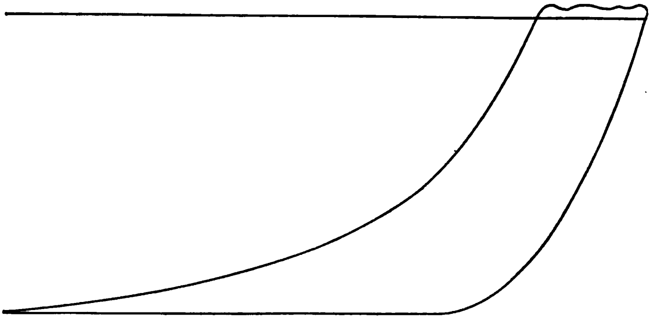


Fig.-16.

The air pump pumps a mixture of water and air. On account of the large clearance the curves are much less steep than in the case of the air compressor.

6. Gas Engine Cards.

Figs. 17 and 18 are from an Otto Gas Engine. Fig. 18 has the firing delayed till the end of the stroke, and shows plainly the effect of not having a "lead" on the firing spark.

Engines working on the Otto cycle can have only one working stroke in four, and as most of these engines work on the "hit and miss" principle, in figuring the horse power from the cards it is necessary to note the actual number of explosions per minute.

Starting at exhaust on the right-hand end, the piston moves to the left, along AB, and drives out the burned gases through the exhaust valve; next the exhaust valve closes, and as the piston moves to the right, gas and air are drawn in, along BA, the mixture being regulated by the opening of the gas inlet, so as to get the proper ratio of gas to air. On the third stroke this mixture is compressed, along AC, and just before the piston reaches the end of the stroke the mixture is fired; the hot gases expand during the fourth stroke, DA.

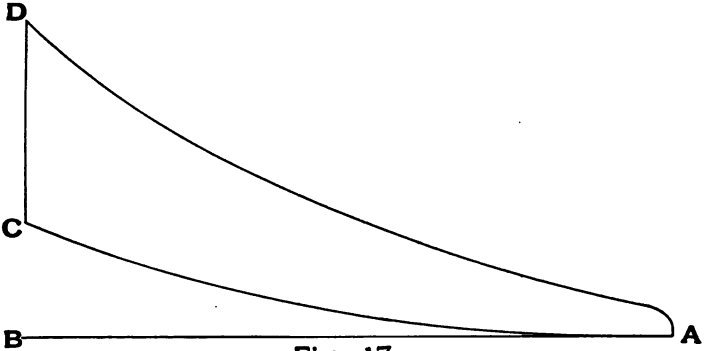


Fig.-17.

If the engine is up to speed and gas is not admitted by the governor, the cylinder is filled with air on the

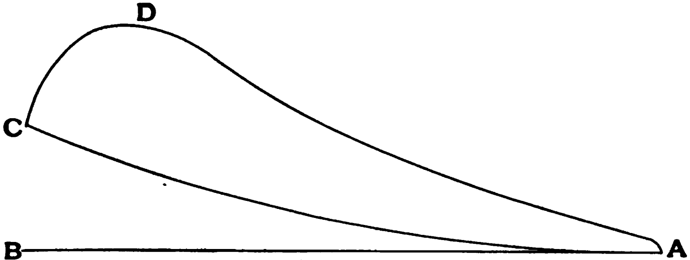


Fig.-18.

filling stroke; this air is compressed on the third stroke and expands back again along nearly the same line on the fourth stroke.

7. Ericsson Hot Air Engine Cards.

Fig. 19 is from an Ericsson hot air engine. This is taken with a 10 lb. spring. As there are no valves in this engine, the same air being alternately heated and cooled, it is impossible to get sharp corners on the card.

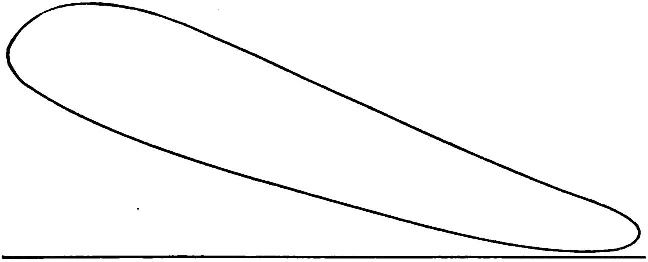


Fig.-19.

8. Stroke Card.

Fig. 20 represents a stroke card from a Corliss engine.

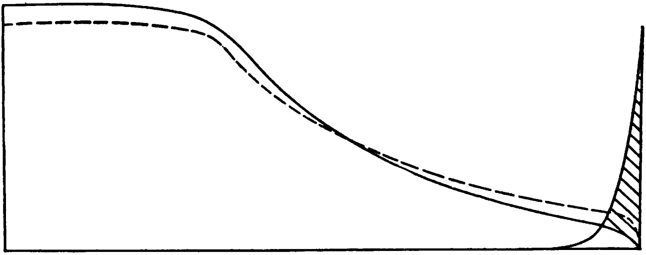


Fig.-20.

This is constructed as explained in connection with Fig. 14.

Near the end of the stroke the pressure on the opposing side of the piston is greater than that on the pushing side, as shown by the part cross-hatched. During this short interval the fly wheel pulls the engine over.

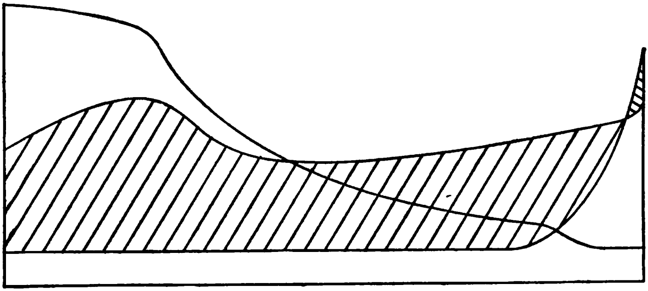


Fig.- 21.

9. Rotative Effect.

The total pressure on the piston rod at any point may be found by multiplying the pressure measured across the stroke card at that point by the area of the piston Suppose the total pressure on the piston rod at the position shown in Fig. 22 is 1000 lbs. Draw the line P to represent this pressure at some assumed scale, say, for example, 1 inch representing 500 lbs. Then P=2″ long. Resolve this along the connecting rod and perpendic-

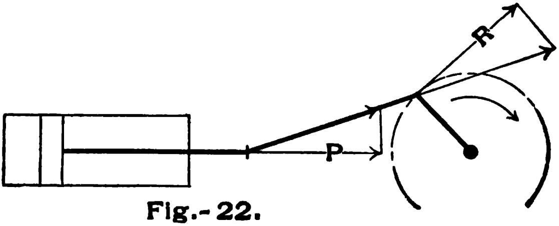


Fig.- 22.

ular to the guides; then the downward thrust at this point on the guides will be the length of this vertical line in inches times 500. It will be noted that the length along the connecting rod is greater than P. Transfer this line in direction and length to the crank pin, and again resolve this into two forces, one, R, at right angles to the crank, and one along the crank. The length R times 500 is the rotative effect, and the short side of the triangle times 500 gives the total compression along the crank.

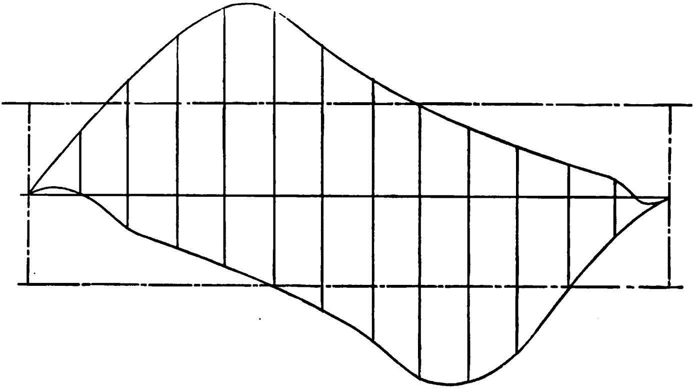


Fig.- 23.

If, now, values of R are calculated for a sufficient number of points and plotted on a line which represents the development of the crank pin circle, a diagram of rotative effect like Fig. 23 is obtained. The upper half comes from one stroke card, and the other half from the other stroke card.

The mean rotative effect is shown by the dot and dash lines.

The amount of energy stored and restored by the fly wheel during a stroke is represented by the area under the curve outside of the dash and dot line.

In the above discussion it has been assumed that the entire pressure on the piston was available for producing rotative effect. This is not the case, however. A certain amount of this pressure is used up at the beginning of the stroke in accelerating the piston, piston-rod, cross-head, and a part of the connecting rod. As these are brought back to rest at the end of the stroke, as much energy is recovered as was lost at the beginning. The dotted line on Fig. 20 shows how the upper line should be changed to make allowance for this, in the case of a low speed Corliss engine. On a high speed engine this is a much larger factor, as may be seen in Fig. 21. In this figure the effective pressure per square inch available for producing rotative effect, is shown by the area cross-hatched.

10. Explosive Force of a Mixture of Gas and Air.

Fig. 24 was obtained with a gas engine indicator and a tuning fork. The mixture, which was one part Boston gas to 6 parts air, was compressed to 25 lbs., as shown at the left of the diagram. It was then fired by a spark, and a pressure of 225 lbs. above the atmosphere resulted. The tuning fork, which was marking on the paper at the same time, gave a means of estimating the time required to reach maximum pressure. Each wave denotes one sixtieth of a second.

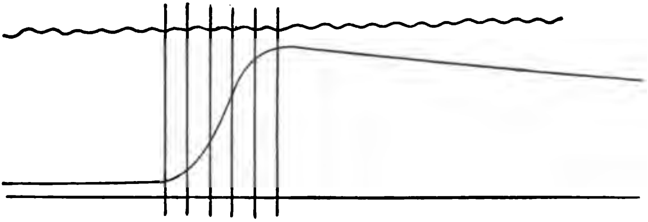


Fig.- 24.

This explosion was $\frac{5}{60}$ or $\frac{1}{12}$ of a second in reaching maximum pressure.

11. Measuring Water Hammer.

Fig. 25 was taken from a 4″ drive pipe 90 feet long, supplying a Rife Hydraulic Ram. The fall of water to the ram was about 8 feet. Beneath the card a wave line drawn by a tuning fork gives a means of measuring time accurately. The maximum pressure is 320 lbs. The recurrence of figures similar to the first, but with rapidly decreasing pressures, seems to indicate a wave traveling back and forth in the pipe. The air which is carried by the water no doubt has something to do with this.

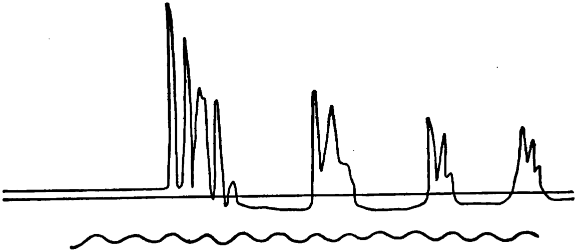


Fig.-25.

12. Flather Dynamometer.

A form of dynamometer built by Mr. John J. Flather makes use of a steam engine indicator for measuring power. Figs. 26 and 27 show the indicator cards taken

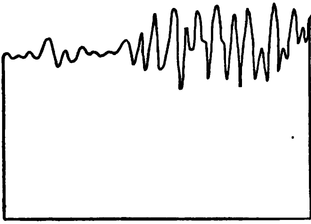


Fig.-26.

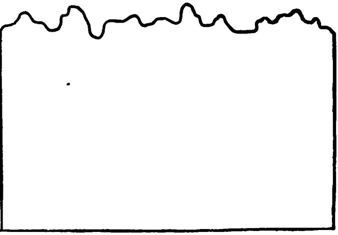


Fig.-27.

from a dynamometer attached to a power drill. Fig. 26 was taken with a sharp drill running into a cored hole in a casting. The hole was coated with rough sand and the drill lost its edge rapidly, when it gave a diagram like Fig. 27.

This dynamometer consists of a hollow shaft on which are two pulleys, one tight and the other loose. On opposite arms of the tight pulley are two small cylinders which are open at one end. These cylinders connect through pipes, at the closed ends, with the hollow shaft. Built out from the arms of the loose pulley are two studs carrying pistons which fit into the cylinders. These cylinders are filled with oil, as is also the hollow shaft, to the end of which the indicator is attached. The loose pulley drives the tight pulley through its pistons, which press on the oil in the cylinders carried by the tight pulley. The steam engine indicator records this pressure. The drum of the indicator is driven from the shaft through a worm and wheel.

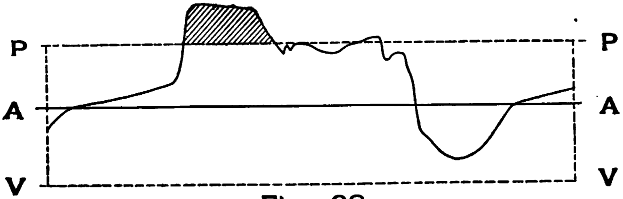


Fig.-28.

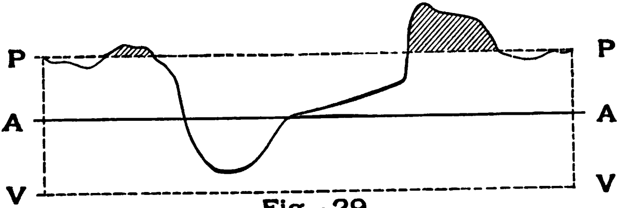


Fig.-29.

13. Pulsometer Steam Pump.

Figs. 28, 29, 30 are taken from the left side, the right side, and the air chamber of a pulsometer steam pump.

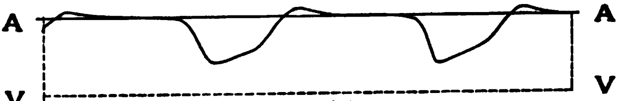


Fig.- 30.

These cards were taken at the same time. The drums of the indicators were moved at the same uniform rate by means of gears driven by a line of shafting. Fig. 31 shows the combined diagram, giving the pressures at any part in the pump at any time.

The line AA represents the atmospheric line, the line VV that corresponding to an absolute vacuum, and

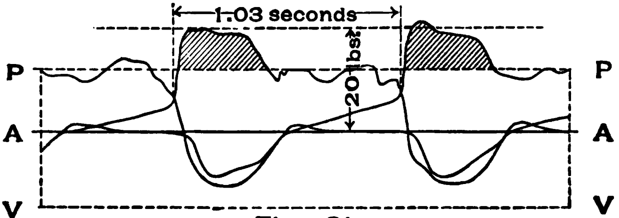


Fig.- 31.

the line PP the pressure in pounds due to the hydrostatic head the pump was delivering against. The shaded portions represent deliveries of water.

PART III.

PART III.

CHAPTER I.

PROPERTIES OF STEAM AND OF PERFECT GASES.

In order to make complete calculations of an engine test, and in order to get as much information as is possible from the cards, it is necessary to understand the properties of saturated steam and to be able to make intelligent use of tables or plots giving such properties.

It is not the intention to give here any lengthy discussion of thermodynamics, and only such parts of that subject will be touched as bear directly on work depending upon the indicator card.

Pressures. The pressure of the atmosphere is usually taken as 14.7 pounds per square inch. This corresponds to a corrected reading on the barometer of 29.92″. Steam gages are made to read pressures above the atmosphere, and to get what is called the *absolute pressure*, the pressure of the atmosphere must be added to that shown by the gage.

A barometric reading in inches of mercury may be reduced to pounds pressure on the square inch by multiplying by .491

To measure pressure below the atmosphere a vacuum gage or a glass U tube filled with mercury may be used.

It is customary to quote the vacuum in inches of mercury instead of in pounds.

If the barometer stood 29.9″ a vacuum of 26″ would mean that there was an absolute pressure of 3.9″, or 3.9 × .491 pounds. All pressures as given in plots or in tables of the properties of saturated steam are absolute pressures.

Temperatures. There are a number of different thermometric scales. The Fahrenheit and the centigrade are the two most generally used.

In the Fahrenheit system melting ice is at 32°, and boiling water or steam at 14.7 lbs. pressure is 212°. In the centigrade system melting ice is 0°, and steam at 14.7 lbs. pressure is 100°. A centigrade reading may be converted into the Fahrenheit scale by multiplying the centigrade reading by $\frac{9}{5}$, and adding 32.

The zero chosen in each of these two systems is purely arbitrary. It is possible to construct a thermodynamic scale of temperature based on thermodynamic laws. The zero of this scale, called the *absolute zero*, is 460.7 degrees below the zero of the Fahrenheit scale and 273.7 degrees below the zero of the centigrade scale.

The absolute temperature at 100° F. is $100+460.7 = 560.7$.

Specific Pressure and *Specific Volume*. Specific pressure is absolute pressure on the square foot. Specific volume is the volume of one pound.

The specific volume of water is $\frac{1}{62.4} = .016$ cubic feet; 62.4 lbs. being the weight of a cubic foot of water at 62°.

The volume of a pound of dry steam represented by the letter s varies with the pressure.

The *British Thermal Unit*, B. T. U., is the amount of heat necessary to raise one pound of water from 62° F. to 63° F.

The *heat of the liquid* is the amount of heat expressed in B. T. U. necessary to raise one pound of water from 32° to the temperature desired.

If the specific heat of water was unity throughout the entire range of temperature, the heat of the liquid would be 32 less than the temperature.

The specific heat is slightly above unity at some temperatures, and slightly below unity at other temperatures.

The heat of the liquid is represented by the letter q.

Relation of pressure and temperature of saturated Steam.

Regnault found that the temperature of steam depended upon the pressure; that the temperature of the steam, if it was not superheated, was exactly the same as that of the water in contact with it.

Particles of water may float in steam the same as fog floats in the air. This does not affect the temperature of the steam.

Total latent heat or *heat of vaporization* (represented by *r*).

If a pound of water at 32° is heated up to the boiling point at atmospheric pressure 180.8 heat units, the value *q* of the heat of the liquid will be consumed in raising the temperature from 32° to 212°. If now heat is added the water will gradually go off as steam. To entirely vaporize this pound 965.8 B. T. U. will be needed. This 965.8 B. T. U. is the heat of vaporization at this pressure. It is often called the total latent heat.

The heat of vaporization is different at different pressures.

The total heat at any pressure (represented by the Greek letter λ called lambda) is the amount of heat necessary to raise a pound of water from 32° to the temperature corresponding to the pressure and to then entirely vaporize the water at this temperature and under the constant pressure.

It is evidently equal to the sum of *q* and *r*, the heat of the liquid and the total latent heat.

Regnault determined by experiment the values of λ, the total heat, at different pressures, and then by subtracting *q*, the heat of the liquid at the temperature corresponding to the pressure, the total latent heat *r* was obtained.

Heat equivalent of external work and *heat equivalent of internal work.*

As water passes into steam at constant pressure and at constant temperature, we have seen that the heat of vaporization *r* is required.

This value *r* is made up of two parts. One part, the heat equivalent of the external work, can be calculated.

The other part, the heat equivalent of the internal work, is obtained by subtracting the first from r.

The following example will illustrate the method of calculating the heat equivalent of the external work.

Suppose one pound of water at 32° to be placed in the bottom of a vertical cylinder of one square foot piston area. Let the piston be weighted so that together with the atmospheric pressure there is a load of 100 pounds per square inch on the piston.

If heat is added to the water, vaporization will not begin till a temperature of 327.58° Fahr. is reached. As vaporization takes place the piston rises in the cylinder.

When all the water has been made into steam the piston will stand 4.403 feet above the bottom of the cylinder.

The pound of water occupied .016 cubic feet, and as the cylinder is one square foot in sectional area, the piston must have moved up a distance of $4.403 - .016 = 4.387$ feet.

The external work done is $100 \times 144 \times 4.387$ foot-pounds.

Dividing this by 778, the mechanical equivalent of one heat-unit, gives as a result 81.2 B. T. U.

The heat equivalent of the external work is represented by $A\,p\,u$. A is the heat equivalent of one foot-pound and is equal to $\frac{1}{778}$. p is the absolute pressure on the square foot. u equals the change in volume in passing from water to steam.

Subtracting the heat equivalent of the external work from the total latent heat, gives the heat equivalent of the internal work. This is represented by the Greek letter ρ (called rho).

In this case at this pressure ρ amounts to

$$884 - 81.2 = 802.8 \text{ B. T. U.}$$

This heat equivalent of the internal work increases as the volume of a pound of steam increases.

As the volume occupied by a pound of steam at very low pressure is large, it will be found that the internal latent heat is a large proportion of the total heat.

Where this heat goes to, may be illustrated thus.

All substances are supposed to be made up of small particles called molecules.

The pound of water occupying .016 cubic feet had a certain number of these molecules. The number remained the same in the pound of steam which filled a volume of 4.403 cubic feet.

The relative distance between these molecules has been increased 275 times.

Each molecule exerts an attraction for its neighbor, and as this attraction has been overcome through space, work has been done. This work requires an equivalent expenditure in heat.

This internal work is often called *disgregation work*.

Total heat
λ

heat of liquid q
heat of vaporization r

heat equivalent of internal work ρ
heat equivalent of external work $A p u$

Volume of a pound of mixture of steam and water.

The volume of a pound of steam is s, the volume of a pound of water is .016 cubic feet.

If the mixture is x parts steam by weight the volume v of the pound of mixture is

$$v = x s + (1 - x) .016$$
$$v = x (s - .016) + .016$$

Total heat of a pound of mixture of steam and water above 32°.

Before any water can be made into steam at a given pressure, the whole of the water must first be heated to the temperature corresponding to the pressure. Then if x parts by weight are made into steam, the heat $x\,r$ must be added, making the total heat to be added $q + x\,r$.

Total heat of a pound of a mixture of steam and water above any given temperature.

First find the heat of the pound of mixture above 32° equal to $q + x\,r$, then subtract the heat of the liquid at the given temperature.

Superheated Steam. Superheated steam is steam of a higher temperature than that corresponding to saturated steam of the same pressure.

The difference of temperature is the number of degrees of superheating.

To tell whether or not steam is superheated a thermometer, a steam gage, and a table or plot giving the temperatures of saturated steam are needed.

Very little is known at the present time about steam of a high degree of superheating.

When the number of degrees of superheat does not exceed 150, the relation between pressure volume and temperature is given very closely by this formula:—

$$pv = 93.5\,T - 971\,p^{\frac{1}{4}}.$$

p is the absolute pressure in pounds on the square foot.

v is the volume of one pound.

T is the absolute temperature of the superheated steam in degrees F. This is found by adding 460.7 to the temperature of the steam as given by the thermometer.

In the last term the pressure is raised to the ¼th power and then multiplied by 971.

Having the temperature and pressure as known terms, the volume may be found, or, with a known volume and a known pressure, the temperature may be found.

Specific Heat of Superheated Steam at Constant Pressure $= C_p$.

It was formerly assumed that this specific heat of superheated steam was a constant for all temperatures. It has recently been shown that the value .48, as determined by Regnault, is not correct, as the specific heat increases with the temperature.

The results of the different experiments made to determine the values of the specific heat at various temperatures are unsatisfactory, but they show that .48 is correct for temperatures near 350° Fahr., and that the value of C_p reaches .75 at temperatures around 600° Fahr.

Total Heat of a Pound of Superheated Steam. This is evidently equal to the total heat of a pound of saturated steam of the same pressure, plus the average value of the specific heat for the range of superheating times the number of degrees of superheat.

Using the letters which represent the different values

$$q + r + C_p \times \text{(degrees superheat).}$$

or

$$\lambda + C_p \times \text{(degrees superheat).}$$

At the back of the book there is a chart which gives the different values of t, q, r, λ, $A\,p\,u$, ρ, and s for pressures from 10 pounds absolute to 250 pounds absolute.

These curves, which are drawn to represent the values given in Peabody's Steam Tables, the tables in general use by engineers, will serve to give values with a moderate degree of accuracy. For accurate work such values should be taken from some reliable steam table which gives these values for each degree difference of temperature or for each pound increase in pressure.

Tables which give values for intervals of 5 pounds, and where values for intermediate points must be obtained by interpolation, are fairly accurate at high pressures, but unreliable at low pressures on account of the error due to interpolation. Even tables reading to 1 pound are unreliable at low pressures for the same reason.

If Peabody's Steam Tables are used, all low-pressure values should be taken from the temperature table which gives the values for each degree from 32°. The pressure corresponding to each temperature is given also. As there are seventy sets of values for pressures from 0 to 1 pound, absolute sufficient accuracy can be obtained.

A table giving the Physical Properties of Saturated Steam Below Atmospheric Pressure appeared in "Power," April, 1902. This table has been reprinted on pages 175, 176.

A table giving the heat of the liquid and the weights of a cubic foot of water for different temperatures between 32° and 212° is given on page 174.

The chart shows that the total heat and the heat equivalent of the external work change but little; that the temperature, the specific volume of steam, and the heat equivalent of the internal work (internal latent heat) change rapidly at low pressures and slowly at high pressures.

In order to show the application of the work given in the preceding pages a few examples will be solved.

(1.) How much heat will it take to make 3 pounds of water at 60° Fahr. into wet steam at 150 pounds absolute pressure? The steam is primed 2 per cent.

If the wet steam contains 2 per cent. moisture there must be 98 per cent. dry steam.

In solving problems in steam where use is made of the values λ, q, r, s, ρ, $A\,p\,u$, etc., it must be remembered that these values are for 1 pound. It is advisable to work all problems as if the actual weight were 1 pound and to finally multiply the result by the actual weight.

The heat in this pound of wet steam above 32° is $q + .98r$ where q and r are the values of the heat of the liquid, and the total latent heat at 150 lbs. absolute, respectively. The water was originally at 60°. The heat of the liquid of water at 60° must be subtracted from this to give the amount to be added per pound.

$$(q_{150\text{ lbs. ab.}} + .98 r_{150\text{ lbs. ab.}} - q_{60^\circ\text{ F.}}) \times 3.$$

From the chart and the table of heat of the liquid these values are

$$(330 + .98 \times 861.2 - 28.1) \times 3 = 3437.6.$$

(2). What volume will the three pounds of wet steam occupy?

From the chart it appears that the volume of one pound of dry steam at 150 lbs. absolute pressure is 3.0 cubic feet.

The volume of one pound of mixture or of wet steam is

$$v = .98\,(3.0 - .016) + .016 = 2.940.$$

The three pounds will occupy a

$$\text{volume} = 3\,v = 8.820 \text{ cubic feet.}$$

(3). An engine is supplied with steam at 144 lbs. absolute pressure. The steam contains 1 per cent. of moisture.

The engine uses 2,800 lbs. of steam per hour (all through the cylinders, there being no jackets).

The indicated horse-power is 200. The temperature of the exhaust at the condenser is 126° F.

The air pump discharges the condensed steam back to the boilers through a primary heater on the exhaust pipe.

The temperature of the feed-water entering the boiler is 100° F.

Each pound of coal burned under the boilers gives up 14,500 B. T. U., 9,900 of which are taken up by the boiler and utilized in making steam.

What is the number of pounds of coal per horse-power as indicated?

What is the thermal unit consumption of the engine per H. P. per minute?

$$(q_{144\text{ lbs.}} + .99 r_{144\text{ lbs.}} - q_{100^\circ\text{ F.}}) \frac{2800}{200}$$

gives the number of thermal units supplied by the boiler per I. H. P. less the heat returned.

Substituting the values from the chart or tables

$$(326.7 + .99 \times 863.5 - 68.0) \times 14 = 15580.$$

Dividing this by 9900 gives the coal per I. H. P. of the engine alone as

$$\frac{15580}{9900} = 1.58 \text{ lbs.}$$

In calculating the thermal unit consumption of the engine it is customary to assume that the condensed steam could be returned by the air-pump to the boiler at the same temperature as that of the exhaust steam.

The thermal unit consumption per I. H. P. per minute is

$$(q_{144\text{ lbs.}} + .99 r_{144\text{ lbs.}} - q_{126^\circ\text{ F.}}) \frac{2800}{200 \times 60} =$$

$$(326.7 + 854.9 - 94.1) \frac{7}{30} = 253.75$$

(4). Suppose that the steam supplied to the engine was of 144 lbs. absolute pressure and 400° F. in temperature; that the steam consumption per hour was 2600 lbs. and that the I. H. P. and other conditions were the same, what would be the B. T. U. per I. H. P. per minute?

$$\left\{ \lambda_{144\text{ lbs.}} + .6\,(400.00 - 355.05) - q_{126^\circ\text{ F.}} \right\} \frac{2600}{200 \times 60} =$$

$$\left\{ 1190.2 + 26.97 - 94.1 \right\} \frac{13}{60} = 243.34$$

The specific heat of superheated steam is taken as .6 at this mean temperature of superheating.

Should the engine be provided with steam jackets the weight of jacket steam per H. P. per minute times $x\,r$ is to be added to the B. T. U. per H. P. per minute through cylinders.

(5.) What is the thermal efficiency of the engines in (4) and (5) as previously explained ?

$$\frac{33000}{778} = 42.42 \text{ B. T. U.}$$

$$\frac{42.42}{253.75} = .167 \qquad \frac{42.42}{243.34} = .174$$

Carnot Engine. It is found in the preceding problem that the thermal efficiency of the engines is low.

One might be led to think that the steam engine was not as economical as it might be made to be. This is not the case, however. Many of our best engines when compared in thermal unit consumption with that of the theoretically perfect engine, working between the same pressures and temperatures, give 70 per cent. comparative efficiency.

The theoretically perfect engine, called the Carnot engine, is not necessarily one with 100 per cent. thermal efficiency, but one in which there are no losses from friction, conduction, radiation, etc.

It is one in which all the heat supplied is accounted for by the sum of the heat withdrawn, and the heat transformed into work.

Evidently an engine to have 100 per cent. thermal efficiency must transform all the heat it receives into work, and have none to throw away or be withdrawn. It can be shown that the efficiency of such a theoretically perfect engine is given by dividing the difference of temperature worked through in the cycle, by the absolute temperature at which heat was supplied to the engine.

A Carnot engine working through the same temperature intervals as those given in examples (4) and (5) would have these thermal efficiencies.

$$\frac{355.05 - 126}{355.05 + 460.7} = .281 \qquad \frac{400 - 126}{400 + 460.7} = .318$$

Comparing the Actual with the Carnot.

$$\frac{.167}{.281} = .59 \qquad \frac{.174}{.318} = .55$$

The thermal unit consumption per H. P. per minute of the Carnot is for the two cases.

$$\frac{42.42}{.281} = 151 \qquad \frac{42.42}{.318} = 133$$

The B. T. U. consumed per H. P. per minute by the actual engine and by the theoretical bear the same ratio as that of the thermal efficiencies.

$$\frac{151}{253 \cdot 75} = \cdot 59 \qquad \frac{133}{243 \cdot 34} = \cdot 55$$

The only correct way to quote the performance of an engine is by its thermal efficiency or by its B. T. U. consumption per I. H. P. per minute.

The weight of steam per H. P. per hour does not mean anything unless one knows the heat in that steam as supplied to the engine and the temperature and pressure of the exhaust.

One engine may develop a H. P. on 9 lbs. of steam, the steam being highly superheated. Another engine with perhaps a higher thermal efficiency than the first may use 12 lbs. per H. P.

If two engines work under exactly the same conditions as to boiler pressure, steam, and vacuum, then a comparison may be made of the steam consumptions per H. P. per hour.

Calculating the Size of a Steam Pipe.

The indicator when applied to the steam chest as explained in Part II. sometimes shows that the steam pipe is not large enough to supply the engine.

If the pipe is furnishing steam to a slow speed engine, and the pipe is not much under the correct size, a drum placed in the steam pipe close to the engine may remedy the trouble. The volume of this drum should be at least four times the volume of the cylinder.

If the pipe supplying a high-speed engine is too small it will have to be changed in order to remedy the trouble.

In figuring the size of a steam pipe or steam main it is customary to allow 6,000 feet velocity of the steam per minute if the pipe is short, with but few elbows.

If the pipe is of moderate length, 5,000 feet per minute, 4,000 is used on long runs where there are many elbows and bends.

Knowing the weight of steam to be carried through the pipe per minute, and knowing also the *lowest pressure* at which the plant will ever work, the volume of the steam can be figured. This volume divided by the allowable velocity will give the area of the pipe needed.

Example: 300 lbs. of steam per minute are to be carried through the pipe. The highest pressure at which the plant runs is 125 lbs. absolute. The lowest 100 lbs. absolute.

From the chart it is found that the volume of one pound of steam at 100 lbs. pressure is 4.4 cubic feet.

$$\frac{4.4 \times 300}{5{,}000} = .264 \text{ sq. ft} = 38.01 \text{ sq. in.}$$

This area corresponds to 6.96″ diameter.

For the higher pressure an area of

$$\frac{3.6 \times 300}{5{,}000} = .216 \text{ sq. ft. is needed.}$$

If the pipe has this area the velocity of steam through the pipe at the lower pressure will be

$$\frac{4.4}{3.6} \times 5{,}000 = 6{,}100 \text{ ft. per min.}$$

Measurement of dry Steam by the Flow through an Orifice.

An empirical formula known as Napiers or as Rankines gives very accurate results.

The orifice should have a rounded edge at entrance.

W = the weight of steam flowing per second.

P_1 = the absolute pressure in pounds per square inch on the entrance side.

P_2 = the absolute pressure in pounds per square inch on the exit side.

A = area of the orifice in square inches.

Where P_1 is equal to or greater than $\frac{5}{3}P_2$

$$W = A\frac{P_1}{70}$$

Where P_1 is less than $\frac{5}{3}P_2$

$$W = A\frac{P_2}{42}\left\{\frac{3(P_1 - P_2)}{2P_2}\right\}^{\frac{1}{2}} = .0292\,A\,(P_1P_2 - P_2^2)^{\frac{1}{2}}$$

As P_2 approaches P_1 more steam goes through the orifice than this formula gives.

This second formula is not to be recommended as accurate within 8 per cent. when $\frac{P_2}{P_1}$ bears the ratio .85 or higher.

Perfect Gases.

The characteristic equation of a perfect gas or the equation giving the relation between the absolute pressure, the volume and the absolute temperature is

$$\frac{Pv}{T} = \frac{P_1v_1}{T_1} = \frac{P_2v_2}{T_2}, \text{ etc.}$$

This relation was determined experimentally.

The volume of a pound of air at 14.7 lbs. pressure and at 492.7° absolute temperature is 12.39 cubic feet; that of hydrogen under the same conditions 178.88 cubic feet.

A few examples will best illustrate how use is to be made of this equation.

1. What will be the volume of one pound of air at 100 lbs. absolute pressure and at 139.3° Fahr.

$$\frac{14.7 \times 12.39}{492.7} = \frac{100 \times v}{460.7 + 139.3}$$

$$v = 2.22$$

2. What will be the weight of a cubic foot of air at this pressure and temperature

$$\frac{1}{v} = \frac{1}{2.22} = .45 \text{ lbs.}$$

3. An air compressor draws in 100 cubic feet of free air per minute at 14.6 lbs. pressure (absolute) and at 60° Fahr. The air is compressed to 200 lbs. absolute and leaves at 120° Fahr. What is the volume of the air discharged?

$$\frac{14.6 \times 100}{460.7 + 60} = \frac{200 \times v}{460.7 + 120} \qquad v = 8.14$$

Measurement of Air by the Flow through an Orifice. Experiments have shown that the following empirical formula gives quite accurate results for orifices up to one inch in diameter.

The orifice should be made with a round entrance, the radius of the curve being equal to the diameter of the orifice, and the length of the straight part of the orifice should be equal to the diameter.

Where the pressure on the entrance side of the orifice is greater than twice the pressure on the exit side

$$w = .530 \frac{P}{T^{\frac{1}{2}}} a$$

w is the weight of air per second. P is the absolute pressure on the square inch on the entrance side. T is the absolute temperature of the air on the entrance side. $a =$ the area of the orifice in square inches.

CHAPTER II.

METHOD OF CALCULATING FROM THE INDICATOR CARD FROM A STEAM ENGINE, THE PER CENT. OF MIXTURE ACCOUNTED FOR AS STEAM AT CUT-OFF AND AT RELEASE.

Draw on the indicator card a line at each end of the card perpendicular to the atmospheric line. Draw a line through the point of cut-off as estimated on the card; also a line just preceding the opening of the exhaust valve as shown on the card, and a line at some point later than the closure of the exhaust valve at compression.

The pressures at cut-off, release, and compression are measured above the atmospheric line and the pressure of the atmosphere added.

To find the per cent. of the stroke at which these events occur, a scale having 100 divisions in a length of 4″ is placed diagonally across the diagram with the zero on the ordinate at one end of the card and the 100 on the ordinate at the other end of the card.

The percentage at cut-off, at release, and at compression can be read directly from this scale.

A certain amount of steam is brought into the cylinder at each stroke, and the same amount by weight is exhausted per stroke.

The weight of steam used per stroke can be found by dividing the total weight of the condensed steam by the total strokes. Call this weight of steam per stroke through the cylinder M.

There is a certain amount of steam in the cylinder at compression. This steam at compression plus the amount brought in per stroke gives the total weight present in the cylinder up to cut-off and up to release. This is called the total weight of mixture.

The weight of steam in the cylinder at compression is found by assuming that the space between the piston and the head of the cylinder, including port passages, is filled with dry steam of the absolute pressure at compression.

The weight at compression is equal to the (per cent. clearance plus the per cent. compression) times the piston displacement and times the weight of a cubic foot of steam at the absolute pressure at compression. The weight of a cubic foot of steam is

$\frac{1}{S}$ or the reciprocal of the volume of a pound.

Call this weight M_0.

The weight of mixture in the cylinder per stroke is $M + M_0$.

The volume at cut-off is equal to the (per cent. clearance plus the per cent. of cut-off) times the piston displacement. Call this V_1.

This volume is filled by $(M + M_0)$ pounds of mixture at the absolute pressure of steam at cut-off as obtained from the diagram.

It has previously been shown that the volume of 1 pound of mixture is $V = X\ (S - .016) + .016$ where X is the per cent. steam by weight.

Hence V_1, the volume of $(M + M_0)$ pounds, must equal $V_1 = (M + M_0)\,X\,(S - .016) + (M + M_0)\ .016$. The volume of 1 pound of steam can be found from tables or from a chart.

X, the only unknown term, is obtained by solving this equation.

The percentage of the mixture accounted for as steam at release is found in a similar way. The volume V_1 is replaced by V_2, the volume including clearance and the piston displacement up to release. S is taken at the absolute pressure at release.

The percentage of mixture accounted for as steam at release frequently is as low as .6.

The indicator by itself does not show that there is any water with the steam in the cylinder. If the steam consumption of an engine is figured from the indicator card, assuming dry steam at release and dry steam at compression, a result as low as 18 pounds may be obtained when the actual consumption, as measured by the steam condensed in the condenser, is as much as 30 pounds.

CHAPTER III.

CALCULATIONS INVOLVED IN ENGINE AND BOILER TESTS.

For the purpose of explaining the necessary calculations indicated above, a compound engine has been selected, as it covers a broader field than one having a single cylinder. The engine chosen is a cross compound, or one in which the cylinders are coupled to the same shaft with cranks at right angles, and is of the Corliss type. As no actual test of this engine has been made all data not derived directly from the diagrams and from the dimensions of the engine have been assumed. The assumptions have been based upon the results obtained from the performance of well constructed engines of this type, and will be found to accord with good practice. In some cases the calculations have been carried to an extent hardly warranted in practice, as the errors of observation would more than counterbalance the omission of many places of decimals. This has been done, however, more as a check on the work, and the degree of accuracy with which such calculations should be done will depend upon circumstances and must be left to the engineer to decide. Only so much of the calculations as is required to determine the ordinary commercial values is given in connection with the boiler test. It will be easy for the engineer to add to this the coal burned and the water evaporated per square foot of heating and grate surfaces, as well as the temperatures and draught pressures in the flues. In every test a full description of *all* the apparatus should be given, and names which in themselves convey no meaning strictly avoided. Smith's patent grate would convey no meaning to any one not familiar with it. The engineer should always avoid expressing an opinion, except where such is beyond all possible doubt, but he should never fail to give all data and give this accurately, so that others may form independent views. Never withhold any data, nor bring any opinion to the front, as the value of many a test has been spoiled from the fact that data which would support particular hobbies have alone been presented. A few hints about the testing of engines and boilers will be given in this article, but the student is strongly advised to continue his work beyond this and consult more extensive treatises.

Too much stress cannot be laid upon the importance of separating the calculations of engine and boiler tests, so that the merits of each can be determined. The rating of an engine by the coal burned is wrong, for the reason that there are too many questions involved. Although a small coal consumption does indicate an economical engine a large one does not necessarily show the reverse. Large coal consumption may result from poor coal, poor firing or other causes entirely disconnected with the engine. The only case in which coal can serve as a basis of measurement of the economy of two or more engines is where each engine is run by the same boiler for quite a length of time under *exactly* the same conditions as to work, coal, weather and care. Nor is it right to take the *weight* of steam used as the basis of measurement, because the steam engine is a *heat* engine and steam of different pressures requires different quantities of heat for its formation. The *true* measure of economy of an engine is the amount of *heat* consumed in the performance of *net* or *useful* work. It is, however, seldom possible to obtain the *net* or useful work done by the engine, and we must, consequently, content ourselves with the indicated work as shown by the diagrams. It will be readily seen that it is of the utmost importance that the greatest care should be observed in taking and computing such diagrams.

To measure the feed-water it should be drawn into a large barrel, containing about two hundred pounds of water, placed upon platform scales, from which barrel it should be drained into another, somewhat larger, to avoid its overflowing, and from this larger barrel the feed pump should take its supply. The outlet pipe of the smaller barrel should be provided with a quick-opening straight-way valve. The weight of the smaller barrel when full and when empty, as well as the difference, should be recorded. It is better in weighing the water not to attempt to get the same amount at each weighing, as a continuous counting of the same number is more liable to result in error. The height of the water in the larger barrel should be accurately determined at the beginning and end of the test. This can be most accurately determined by driving a nail through the side of the barrel and so regulating the water drawn from the barrel, by means of a by-pass in the discharge pipe of the feed pump, that the water at the beginning and the end shall just reach the nail. It can also be done by immersing a stick in the barrel and noting the height of the water at the beginning and end of the test and adding or deducting the difference, as the case may be, from the total amount of water weighed. The water can be brought to the *same* height on the stick at the beginning and end of the test as before. The advantage of the nail is the fact that it remains and no doubt can exist as to the height. It is important that the height of the water in the boiler should be exactly the same at the end as at the beginning of the test, as in that case all changes which have occurred during the intervals are due to the water used. It is even well to keep the height of the water in the boiler nearly the same throughout the trial. The height of the water can be marked on a little wooden stick attached to the water glass.

In measuring the water drained from the jackets extra care should be observed. A good device is to take a 4-inch wrought iron pipe, about twelve inches long, and cap each end. Connect the top with a pipe leading to the jacket, in which pipe insert a straight-way valve. Into the bottom screw a pipe leading to the weighing barrel, and in this pipe, close to the 4-inch pipe, insert a *globe* valve, to the wheel of which attach a long wooden stick. This arrangement enables the attendant to easily regulate the flow from the apparatus. Attach a water glass to the side of the apparatus. The height of the water in the glass, and, consequently, the rate of flow, should be kept constant throughout the trial, so that the amount wasted during the emptying and replenishing of the weighing barrel with cold water, can be accurately determined. The weighing barrel should contain cold water, and the moment vapor begins to appear it should be emptied and replenished with cold water, because otherwise water would be lost in the escaping vapor and the result be inaccurate. The barrel should be placed upon platform scales accurately tested. The apparatus for regulating the flow of water from the jackets should be well covered with some good non-conducting material, and an accurate thermometer should be placed in an oil cup in the supply pipe. Use none but the most accurate thermometers, as there is a great difference between those of various makers.

BOILER TEST.

Some engineers maintain that it is impossible to determine whether the fires at the end of a test are in the same condition as they were in at the beginning, and, therefore, recommend that the fires be always drawn at the beginning and end. On the other hand, many claim that drawing the fires at the beginning allows too great a loss of heat, which objection, however, the former contend is more than offset by greater accuracy in determining the amount of coal used. The shorter the test the more important the determination of the amount of coal used.

In any case the condition of the walls, pressure of steam, and everything about the boiler just before beginning should be exactly the same as that which is to obtain during the test.

If the fires are to be drawn, the above rules being observed, about five or ten minutes before the test is to commence the valve in the steam pipe from the boiler should be closed and the steam shut into the boiler, and the pressure and height of the water carefully noted. The fires should then be drawn as rapidly as possible and the grates and ash pit thoroughly cleaned. Without any unnecessary loss of time wood and coal should be put upon the grates and new fires started. This should occur as nearly as possible at the time appointed for starting. The steam pressure should be noted and the water in the glass marked and the boiler test should then be considered as begun. All coal and wood not used in starting the fires should be weighed back. As soon as the steam in the boiler has reached the pressure, which is to obtain during the engine test, the valve should be opened and the engine started, the time being carefully noted. A few minutes before closing the test the steam should be shut into the boiler and the engine allowed to run until it stops, at which time the engine test is to be considered as ended, and observations should be made at this time. The fires should now be drawn and when this is done the boiler test is ended. Care must be taken that the level of the water in the boiler is kept the same at the end as at the beginning of each test, that is to say, the water in the glass should be at the same height at the end as at the beginning of the boiler test and at the same height at the end as at the beginning of the engine test. Should the water be lower at the end of the test it can be raised to the same level after the test and the amount required carefully weighed. This amount must be considered as forming part of the quantity used. All the water fed to the boiler should be carefully weighed and the temperature at its entrance to the boiler carefully noted.

For fuller details regarding the testing of boilers the reader is referred to Vol. VI., 1884–1885 of the "Transactions of the American Society of Mechanical Engineers."

Dimensions of the Engine.

The dimensions of the engine on which the test described in the following pages was made are:

Diameter of the high pressure cylinder	30 inches
Diameter of the low pressure cylinder .	56 inches
Stroke of both pistons	72 inches
Diameter of the piston-rod of the high pressure cylinder	$4\frac{7}{8}$ inches
Diameter of the piston-rod of the low pressure cylinder	$5\frac{1}{4}$ inches
Diameter of vertical single acting air-pump	39 inches
Stroke of same	15 inches
Volume of clearance space in each end of cylinder: —	
High pressure cylinder .	1611 cubic inches
Low pressure cylinder .	5884 cubic inches

CALCULATIONS FOR ENGINE AND BOILER TESTS.

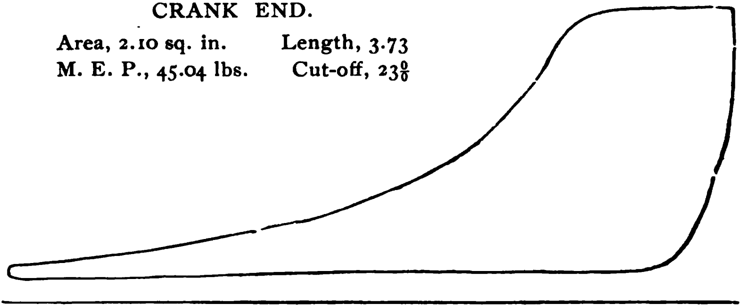
DIAGRAMS FROM THE HIGH PRESSURE CYLINDER.

March 17, 1893.

Diameter of cylinder, 30 inches; stroke, 72 inches; piston-rod, 4⅞ inches. Throttle wide open. Barometer 30.1 in. at 75.96°F., or 29.9 inches at 32°F. Atmospheric pressure, 14.7 lbs. Average pressure by gage on boiler, 124.0 lbs. Average pressure by gage at throttle, 122. Average vacuum by gage on condenser, 27.3 inches. Temperature of feed at boiler, 111°F. Revolutions per minute, 58. Scale of diagrams, 80. Diagrams No. 63.

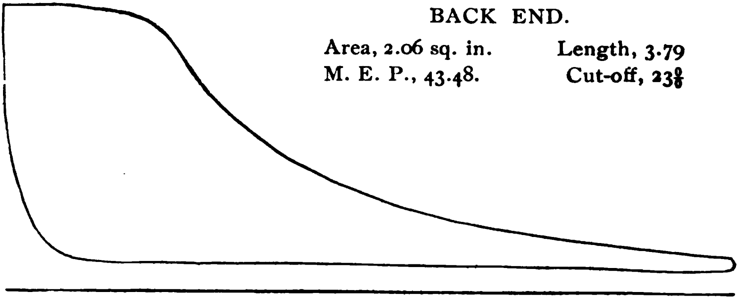
CRANK END.

Area, 2.10 sq. in. Length, 3.73
M. E. P., 45.04 lbs. Cut-off, 23%



BACK END.

Area, 2.06 sq. in. Length, 3.79
M. E. P., 43.48. Cut-off, 23%



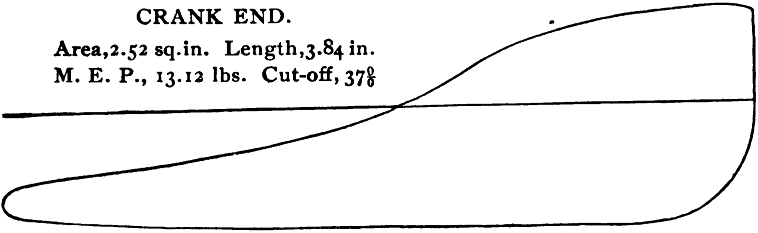
DIAGRAMS FROM THE LOW PRESSURE CYLINDER.

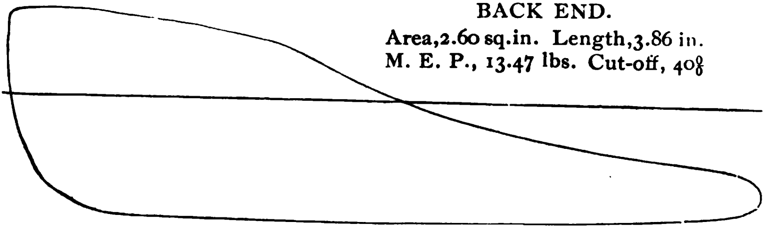
March 17, 1893.

Diameter of cylinder, 56 inches; stroke, 72 inches; piston rod, 5¼ inches. Throttle wide open. Barometer, 30.1 inches at 75.96°F., or 29 9 at 32°F. Atmospheric pressure, 14.7 lbs. Average pressure by gage on boiler, 124.0 lbs. Average pressure by gage at throttle, 122 lbs. Average vacuum by gage on condenser, 27.3″. Temperature of main feed at boiler, 111°F. Revolutions per minute, 58. Scale of diagrams, 20. Diagrams No. 63.

CRANK END.

Area, 2.52 sq.in. Length, 3.84 in.
M. E. P., 13.12 lbs. Cut-off, 37%





BACK END.

Area, 2.60 sq.in. Length, 3.86 in.
M. E. P., 13.47 lbs. Cut-off, 40%

DETERMINATION OF THE ATMOSPHERIC PRESSURE.

Readings of the barometer.

Time.	Temp. shown by attached thermometer.	Reading of height.
10 A. M.	74.45°F.	29.9 inches.
3 P. M.	77.47°F.	30.3 inches.
Sum,	151.92	60.2
Means,	75.96°F.	30.1 inches.

Reduction to the freezing point.

$$\text{Height} = \frac{\{1 + s(t - 62)\}\, h}{1 + m(t - 32)}$$

where

h = reading of the barometer.

t = temperature of the attached thermometer.

m = expansion of mercury for 1° F., taken as 0.0001010 of its length at 32° F.

s = expansion of the substance of which the scale is made; for brass s is taken as 0.00001020 of its length (h) at the standard temperature for the scale, viz: 62° F.

See: "Instructions for Voluntary Observers," U. S. Department of Agriculture, page 39, Washington, D. C., 1892.

Substituting the above values in the formula we have:—

$$\text{Reduced height} = \frac{\left\{1 + 0.00001020(75.96 - 62)\right\} \times 30.1}{1 + 0.0001010\,(75.96 - 32)} = 29.97$$

The weight of one cubic inch of mercury of the temperature of 32° F., is 0.491 pound and, consequently, the pressure of the atmosphere will be $29.97 \times 0.491 = 14.7$ pounds per square inch.

Be sure and take the temperature first as the heat of the observer's person affects the thermometer very quickly.

CALORIMETRIC TEST.

Time.	Gage on Steam Pipe.	Gage on Calorimeter.	Temp. in Calorimeter	Remarks.
1 10.00 A. M.	121.75	p_1	t_s	Calorimeter opened.
	p			
2 10.15	122.00	11.4	302.1° F.	
3 10.45	122.25	11.4	302.1° F.	
4 11.00	122.00	11.3	302.1° F.	Calorimeter closed.
Totals	366.25	34.1	906.3° F.	
Means	122.08	11.36	302.1° F.	
Atmospheric pressure	$14.7 = p_a$	$14.7 = p_a$		
Absolute pressure	136.78	26.06		

Gage pressure for flow of steam for one hour (average of four readings above) = $122.0 = p$

Absolute pressure = $122.0 + 14.7$ or 136.7 lbs.

The weight of steam, in pounds, flowing through the orifice per second will be: (diameter of orifice 0.2 of an inch.)

$$\frac{(p + p_a) \times \text{area of orifice in inches}}{70} = \frac{(122 + 14.7) \times 0.031415}{70}$$

and for one hour,—

$$\frac{3600 \times 136.7 \times 0.031415}{70} = 220.9 \text{ pounds.}$$

The weight of steam in one pound of mixture of steam and water will be,—

$$x = \frac{\lambda_1 + c_p\,(t_s - t_1) - q}{r}$$

where

p_a = atmospheric pressure in pounds per square inch.
p = gage pressure in steam pipe.
r = heat of vaporization for $p_a + p$.
q = heat of liquid for $p_a + p$.
p_1 = gage pressure of steam in calorimeter.
t_s = temperature of steam in calorimeter.
t_1 = temperature of steam corresponding to $p_a + p_1$.
λ_1 = total heat of steam, reckoned from 32° F., corresponding to $p_a + p_1$.
c_p = specific heat of superheated steam of constant pressure = 0.48.

For the present case we have,—

$t_s = 302.1^\circ$ F. $p = 122.08$ pounds. $p_1 = 11.36$ pounds
$p_a = 14.7$ pounds. $p_a + p = 136.78$ pounds. $p_a + p = 26.06$ pounds.

From Professor Peabody's steam tables we have:—

For pressure 136 lbs. $r = 866.9$
" " 137 lbs. $r = 866.4$
" " 136.78 lbs. $r = 866.9 - 0.78\,(866.9 - 866.4)$
$= 866.9 - (0.78 \times 0.5) = 866.5$

For pressure 136 lbs. $q = 322$
" " 137 lbs. $q = 322.6$
" " 136.78 lbs. $q = 322 + 0.78\,(322.6 - 322)$
$= 322 + (0.78 \times 0.6) = 322.47$

For pressure 27 lbs. $t = 244.32$
" " 26 lbs. $t = 242.21$
" " 26.06 lbs. $t_1 = 242.21 + 0.06\,(244.32 - 242.21)$
$= 242.21 + (0.06 \times 2.11) = 242.3$

For pressure 27 lbs. $\lambda = 1156.5$
" " 26 lbs. $\lambda = 1155.8$
" " 26.06 lbs. $\lambda_1 = 1155.8 + 0.06\,(1156.5 - 1155.8)$
$= 1155.8 + (0.06 \times 0.7) = 1155.84$

$$x = \frac{\overset{\lambda_1}{1155.84} + \overset{c_p}{0.48}\,(\overset{t_s}{302.1} - \overset{t_1}{242.3}) - \overset{q}{322.5}}{\underset{r}{866.5}} = 0.995 \text{ of a pound.}$$

and the weight of moisture per pound of mixture $1 - 0.995 = 0.005$ of a pound. That is to say, the steam contains one-half of one per cent. of moisture.

BOILER TEST.

Duration of test,	12 hours 30 minutes.
Pressures in pounds per sq. inch,	
Of the atmosphere,	14.7
Steam as shown by gage,	124.0
Steam on boiler above zero,	138.7
Temperatures in degrees Fahrenheit,	
Average of the feed water at its entrance to the boiler,	111°
Average of the jacket drain at its entrance to the boiler,	295°

Total heat, reckoned from 32° F. of steam of pressure,
of 138 lbs. $= \lambda = 1189.2$ B. T. U.
of 139 lbs. $= \lambda = 1189.4$ B. T. U.
of 138.7 lbs. $= \lambda = 1189.34$ B. T. U.

Heat of liquid corresponding to temperature,

111° F. = temperature of feed water,	79 B. T. U.
1189.34 — 79 =	1110.34 B. T. U,

or the number of British thermal units given to every pound of water fed to the boiler, under the assumption that all of this water was converted into steam of boiler pressure.

COAL AND ASHES.

Wood put upon the grates at the beginning,	434.75 pounds.
Coal equivalent = 434.75 × 0.40 =	173.9 pounds.
Total amount of coal put upon the grates,	25,694.0 pounds.
Coal and coal equivalent of wood,	25,867.9 pounds.

Refuse at end and during test.

From grates at end of test,	183 pounds.
From ash pit at end of test,	157 pounds.
From ash pit during test,	2252 pounds.

Analysis of refuse,

	At end of test. from grate.	 from pit.	During test. from pit.
Coal, . .	5¼ oz.	3⅛ oz.	3 oz.
Ash and cinders, .	7¾ oz.	7⅞ oz.	21 oz.
Weight analyzed,	13 oz.	11 oz.	24 oz.

Proportion and Weight of coal in refuse,

	Proportion of	Weight of
From grates at end,	21 to 52	$\frac{21}{52}$ of 183 = 73.9 lbs.
From pit at end,	25 to 88	$\frac{25}{88}$ of 157 = 44.6 lbs.
From pit during test,	3 to 24	$\frac{1}{8}$ of 2252 = 281.5 lbs.
Total weight of unburnt coal in refuse, .		400.0 lbs.

Proportion and weight of ash and cinders in refuse,

	Proportion of	Weight of
From grates at end,	31 to 52	$\frac{31}{52}$ of 183 = 109.09 lbs.
From pit at end,	63 to 88	$\frac{63}{88}$ of 157 = 112.39 lbs.
From pit during test,	7 to 8	$\frac{7}{8}$ of 2252 = 1970.50 lbs.
Total weight of ash and cinders in refuse,		2191.98 lbs.

Coal burned = 25,867.9 — 73.9 = . . 25,794 pounds.

Per cent. of ash and cinders in coal,

(100 × 2191.98) ÷ (25,694 — 400) = . . . 8.66

Total water fed to the boiler, . . . 238,235 pounds.

Ash and cinders per pound of coal, . 0.0866 of a pound.

Combustible per pound of coal, . . 0.9134 of a pound.

Water apparently evaporated,

A. From temperature of feed, 111° F., into steam of boiler pressure, 138.7 lbs. per square inch above zero,

1st : per pound of coal burned 238,235 ÷ 25,794 = 9.236 lbs.

2nd : per pound of combustible 9.236 ÷ 0.9134 = 10.11 lbs.

B. Had feed water been supplied at the temperature of 212° F. and converted into steam of the same temperature.

1st : per pound of coal (1110.34 × 9.236) ÷ 965. 8 = 10.618 lbs.

2nd : per pound of combustible 10.6180 ÷ .9134 = 11.624 lbs.

Note. 965.8 are the number of B. T. U. required to convert *one pound* of water of the temperature of 212° F. into steam of the same temperature.

Heat per pound of coal = 9.236 × 1110.34 = 10,255.1 B. T. U.

In making a boiler test some deduct the unburnt coal found in the ash pit, as well as that taken from the grate, from the total coal put upon the grate and consider the result as the coal *burned* during the trial. This may and may not be fair, for the reason that most of the coal which has fallen through the grates is commercially of no value and is usually thrown away with the refuse in the pit. If the grates are faulty in construction and allow an excessive amount of coal to fall through them, although the fault should be remedied, the boiler, when considering the evaporation, should not be held responsible, if it is all right in other respects. Strictly speaking *all* unburnt coal should be deducted from that put upon the grates, if it is desired to accurately determine the calorific value of the coal aside from any commercial considerations. These are questions which must be left to the engineer to decide in each case, but it has been thought best to call attention to them.

COAL USED DURING ENGINE TEST.

Average pressure in the boiler per sq. in. above zero,	138.7 lbs.
Average temperature of main feed water at its entrance to the boiler,	111° F.
Average temperature of the water drained from the jackets at its entrance to the boiler when the engine is running in its regular way,	295° F.
Total heat, reckoned from 32° F., given to each pound of steam of boiler pressure (see boiler test, page 68).	1189.34 B. T. U.
Heat, reckoned from 32° F., of the main feed water (temperature 111° F.),	79. B. T. U.
Heat imparted to each pound of mixture of steam and water passing through the cylinders of the engine 1189.34 — 79 =	1110.34 B. T. U.
Heat, reckoned from 32° F., in one pound of water fed to the boiler from the jackets when the engine is running in its regular way (temperature 295°),	264.5 B. T. U.
Heat imparted to each pound of mixture of steam and water passing through the jacket 1189.34 — 264.5 =	924.84 B. T. U.
Total weight of mixture of steam and water passing through the cylinders of the engine,	216,492 pounds
Total weight of the feed water consumed by the jackets of the engine	15,369.7 pounds.

The total amount of heat given to the feed water consumed by the cylinders and jackets will be:—

By the cylinders 1110.34 × 216,492 = 240,379,727.28 B. T. U.
By the jackets 924.84 × 15,369.7 = 14,214,513.348 B. T. U.
By the cylinders and jackets, 254,594,240.628 B. T. U.

The heat imparted to the feed water by each pound of coal burned (see boiler test) = 10,255.1 B. T. U.

The weight of coal burned during the engine test will then be:—

254,594,240.628 divided by 10,255.1 = 24,826.11 pounds.
Coal per hour, 24,826.11 ÷ 12 = 2,068.84 pounds.
Coal per horse power per hour 2068.84 ÷ 1338.617 = ... 1.54 pounds.

HORSE POWERS.

High pressure steam cylinder.

Diameter of cylinder,	30 inches.
Length of stroke 6 feet or,	72 inches.
Diameter of piston-rod,	4⅞ inches.
Area of circle 30 inches in diameter,	706.86 sq. in.
Area of circle 4⅞ inches in diameter,	18.665 sq. in.
Net area of piston in square inches,	
Crank-end 706.86 — 18.665,	688.195
Back-end	706.86
Length of diagrams, in inches,	
Crank-end,	3.73
Back-end,	3.79
Area of Diagrams, in square inches,	
Crank-end,	2.10
Back-end,	2.06
Mean width of diagrams, in inches,	
Crank-end, 2.10 ÷ 3.73,	0.563
Back-end, 2.06 ÷ 3.79,	0.5435
Scale of diagrams,	80
Mean effective pressures per square inch,	
Crank-end 80 × 0.563	45.04 pounds.
Back-end 80 × 0.5435	43.48 pounds.
Average number of revolutions per minute,	58

The horse power of the high pressure cylinder will be,—

$$\frac{(\{688.195 \times 45.04\} + \{706.86 \times 43.48\}) \times 58 \times 6}{33{,}000} = 650.976$$

Approximate method;

Average mean effective pressure of both ends=	
$\frac{1}{2}(45.04 + 43.48)$	44.26 pounds.
Average net area of piston at both ends =	
$\frac{1}{2}(688.195 + 706.86)$	697.528 sq. in.

Horse power,—

$$\frac{697.528 \times 44.26 \times 2 \times 58 \times 6}{33{,}000} =$$

$= 0.2536 \times 44.26 \times 58 =$	651.0114

Low pressure steam cylinder.

Diameter of cylinder,	56 inches.
Length of stroke, 6 feet, or	72 inches.
Diameter of piston rod,	5¼ inches.
Area of circle 56 inches in diameter,	2,463.01 sq. in.
Area of circle 5¼ inches in diameter,	21.648 sq. in.
Net area of piston in square inches,	
Crank-end, 2,463.01 — 21.648,	2,441.362
Back-end,	2,463.01
Length of diagrams, in inches,	
Crank-end,	3.84
Back-end,	3.86
Area of diagrams, in square inches,	
Crank-end,	2.52
Back-end,	2.60
Mean width of diagrams, in inches,	
Crank-end,	0.6562
Back-end	0.6735
Scale of diagrams	20
Mean effective pressures, per square inch,	
Crank-end, 20 × 0.6562	13.12 pounds.
Back-end, 20 × 0.6735	13.47 pounds.
Average number of revolutions per minute,	58

The horse power of the low pressure cylinder will be,

$$\frac{(\{2{,}441.362 \times 13.12\} + \{2{,}463.01 \times 13.47\}) \times 58 \times 6}{33{,}000} = 687.641$$

Approximate method,

Average mean effective pressure of both ends	
$= \frac{1}{2}(13.12 + 13.47) =$	13.295 pounds.
Average net area of piston at both ends =	
$\frac{1}{2}(2{,}441.362 + 2{,}463.01) =$	2,452.186 sq in.

Horse power,

$$\frac{2{,}452.186 \times 13.295 \times 2 \times 58 \times 6}{33{,}000} = \quad . \qquad 687.602$$

Horse power of the whole engine. Accurate method.

Horse power of high pressure cylinder, .	650.976
Horse power of low pressure cylinder, .	687.641
Horse power of the whole engine, . .	1,338.617
Ratio of horse powers, ...	
Power of high pressure cylinder to that of low pressure cylinder 650.976 to 687.641 .	0.946
Power of low pressure cylinder to that of high pressure cylinder 687.641 to 650.976 .	1.05
Power of high pressure cylinder to that of whole engine 650.976 to 1,338.617 .	0.486
Power of low pressure cylinder to that of whole engine 687.641 to 1,338.617 .	0.513

HORSE POWER CONSTANT.

Where the horse power is to be frequently calculated it is best to reduce the numerical work by finding the engine's *constant*. In the calculation for the horse power of the high pressure cylinder,—

$$\frac{\{(688.195 \times 45.04) + (706.86 \times 43.48)\} \times 58 \times 6}{33{,}000}$$

the quantities 688.195, 706.86, 6 and 33,000 are the same no matter at what speed and under what pressure the engine runs. Consequently dividing through by the area 706.86 square inches we have the above in the form,—

$$\left\{\frac{(688.195 \times 45.04)}{706.86} + 43.48\right\}\frac{\times 706.86 \times 58 \times 6}{33{,}000}$$

But,

$$\frac{688.195}{706.86} = 0.9736 \quad \text{and}$$

$$\frac{706.86 \times 6}{33{,}000} = 0.12852$$

and the horse power will then be,

$$\{(0.9736 \times 45.04) + 43.48\} \times 0.12852 \times 58$$

or $(43.85 + 43.48) \times 0.12852 \times 58$

$$= 87.33 \times 0.12{,}852 \times 58 = \qquad . \quad 650.97$$

The quantities which vary in the above are: 45.04, the mean effective pressure of the crank end; 43.48, that of the back end; and 58, the number of revolutions per minute.

The constant, using the *approximate* method of figuring the horse power is found thus:

Horse power of high

$$\frac{697.528 \times 44.26 \times 2 \times 58 \times 6}{33{,}000}$$

here all the quantities but 44.26, the mean effective pressure of both ends, and 58, the number of revolutions per minute, are constant.

consequently $\frac{697.528 \times 2 \times 6}{33{,}000} = 0.2536$ is the constant sought and the horse power will be

$$0.2536 \times 44.26 \times 58 = 651.0114$$

Power of Engine for One Stroke of the Piston.

In the investigation of the steam engine it is sometimes necessary to obtain the power for one stroke of the piston. In such a case we must take the pressure urging the piston forward from one diagram, and that resisting its movement from the diagram of the opposite end of the cylinder.

EXAMPLE.

The power urging the piston *forward* must be found from the area obtained by measuring the diagram from the *upper* down to the zero line and the resistance opposing its movement must be determined from the area included between the *lower* and *zero* line of the diagram from the opposite end of the cylinder.

Area of diagram from crank end of high pressure cylinder included between the upper and zero lines	3.47 sq. in.
Mean width corresponding, 3.47 ÷ 3.73 = .	0.93 inch.
Mean effective pressure corresponding 0.93 × 80 =	74.4 lbs.
Area of diagram from back end of high pressure cylinder. included between *lower* and zero lines, ...	1.36 sq. in.
Mean width corresponding 1.36 ÷ 3.79 .	0.3588 sq. in.
Mean effective pressure corresponding=0.3588×80 =	28.70 lbs.

The *net* power urging the piston on the crank side forward will then be:—

$$(\overset{\text{area.}}{688.195} \times \overset{\text{press.}}{74.4}) - (\overset{\text{area.}}{706.86} \times \overset{\text{press.}}{28.7}) = 30{,}914.826 \text{ pounds.}$$

It will be unnecessary to make the calculation for the other end, as it was only intended to show the principle.

Weight of steam used per horse power per hour as determined from the weight of water fed to the boiler.

Total water fed to the boiler during 12 hours,	231,861.7 pounds.
Weight of feed water per hour,	19,321.8 pounds.
Weight of steam per horse power per hour = 19,321.8 ÷ 1,338.617	14.43 pounds.

HEAT USED BY THE ENGINE.

Atmospheric pressure in pounds per sq. inch,	14.7
Average pressure of steam, by gage, near the engine	122.0 pounds.
Average pressure of steam in pounds per square inch above zero, ...	136.7
Temperature of feed water near engine,	114° F.
Temperature of water drained from jackets taken near the engine, . . .	338° F.
Total water fed to the boiler during engine test and apparently evaporated into steam of boiler pressure, ...	232,082.6 lbs.
Total weight of water passing through the calorimeter, ...	220.9 lbs.
Net amount of mixture of water and steam passing through the engine, . .	231,861.7 lbs.
Weight of water drained from the jackets, .	15,369.7 lbs.
Weight of mixture passing through the cylinders, ...	216,492.0 lbs.

From Professor Peabody's steam tables we have:—

For pressure = 136 $r = 866.9$ B. T. U.; $q =$ 322. B. T. U.

137 $r = 866.4$ B. T. U.; $q =$ 322.6 B. T. U.

Consequently for pressure=136.7 r=866.5 B. T. U.; q=322.4 B. T. U.

For temperature of feed 114° $q_0 =$.	82 B. T. U.
For temperature of jacket drain 338° $q_1 =$.	308.8 B. T. U.
Weight of dry steam per pound of mixture of steam and water delivered to the cylinders and jackets, as determined by the calorimetric test, ...	0.995 lb.
Heat required to raise one pound of water from the temperature of 32° F. to that corresponding to steam of 136.7 pounds per square inch above zero = q= . .	322.4 B. T. U.

Heat required to convert one pound of water of the temperature corresponding to that of steam of 136.7 pounds per square inch above zero, into steam of this pressure = r =	866.5 B. T. U.
The heat, reckoned from 32° F., imparted to each pound of mixture of steam and water delivered to the cylinders and jackets will, consequently, be distributed as follows:—	
Heat imparted to the water, . . .	322.4 B. T. U.
Heat imparted to the steam, = 0.995 × 866.5 =	862.1 B. T. U.
Heat imparted to the mixture, reckoned from 32° F.,	1184.50 B. T. U.
Each pound of water used to form this mixture was pumped through a coil heater in the exhaust pipe of the low pressure cylinder, leaving this heater at a temperature of 114° F., and consequently carrying to the boiler a quantity of heat, reckoned from 32° F. of q_0 =	82 B. T. U.
Therefore the difference between the heat received and rejected, or returned to the boiler, represents the amount of heat utilized in each pound of mixture delivered to the cylinders; this was 1184.5 — 82 = .	1102.5 B. T. U.
As each pound of mixture delivered to the jackets carried with it the same amount of heat as that carried by each pound entering the cylinder, or 1184.5 B. T. U., reckoned from 32° F., and as each pound discharged from the jackets had a temperature of 338° F., or 308.8 B. T. U., reckoned from 32° F., the amount of heat utilized in each pound of mixture delivered to the jackets will be: 1184.5 — 308.8, or .	875.7 B. T. U.
Total weight of mixture delivered to cylinder,	216,492 lbs.
Total weight of mixture delivered to jackets .	15,369.7 lbs.
The total amount of heat utilized will then be,	
In the cylinders 216,492 × 1102.5 =	238,682,430.0 B. T. U.
In the jackets 15,369.7 × 875.7 =	13,459,246.29 B. T. U.
In the cylinders and jackets,	252,141,676.29 B. T. U.
Duration of engine test 12 hours or . .	720 minutes.
Heat utilized per min.=252,141,676.29÷720=	350,196.77 B. T. U.

Heat per horse power per minute 350,196.77 ÷
1338.617 = ... 261.61 B. T. U.

Each horse power is equal to 33,000 foot pounds per minute, and as each thermal unit is equivalent to 778 foot pounds, the quantity of heat which would be utilized per horse power per minute, could all this heat be converted into useful work, would be: 33,000 ÷ 778 or 42.42 B. T. U.

The efficiency of the engine is then
(100 × 42.42) ÷ 261.61 or . . . $16\frac{21}{100}$ per cent.

STEAM CONSUMPTION AS DETERMINED FROM THE INDICATOR DIAGRAMS OF HIGH PRESSURE STEAM CYLINDERS.

No reliable results can ever be obtained of the weight of steam *actually* consumed by the engine from Indicator Diagrams. The only legitimate use which can be made of diagrams for the purpose of measuring the weight of steam, is simply as auxiliary in the investigation of the steam engine, where the actual weight and quality of the steam have been determined by the most careful measurements.

It is, however, necessary for the engineer to know and understand the various methods employed for making such calculations, and they are given in the following pages.

CASE I. WHERE THE CLEARANCE SPACE IS KNOWN.

Net area of piston in square inches,
Crank-end 706.86 — 18.665 = . . 688.195
Back end, ... 706.86
Stroke = 72 inches or . . . 6 feet.
Piston displacement per stroke,
Crank-end = { 688.195 × 72 } ÷ 1728 = 28.675 cubic feet.
Back-end = { 706.86 × 72 } ÷ 1728 = 29.453 cubic feet.
Clearance space,
Crank-end, 1611 cubic inches or 0.932 cubic foot.
In fraction of piston displacement
1611 ÷ { 688.195 × 72 } = . . 0.0325
Back-end, 1116 cubic inches or . . 0.932 cubic foot.
In fraction of piston displacement
1611 ÷ { 706.86 × 72 } = . . . 0.0316
Volume of cylinder in cubic feet,
Crank-end, 28.675 + 0.932 = . . 29.607
Back-end, 29.453 + 0.932 = . . 30.385

Length of Diagrams in inches,

Crank-end,	3.73
Back-end,	3.79

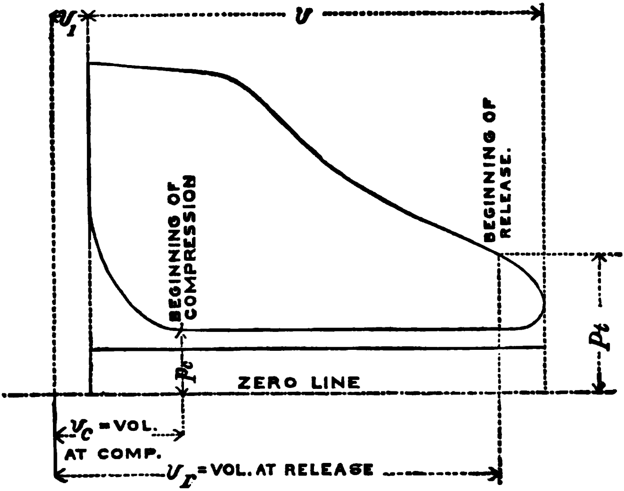


FIG. 21.

Distance travelled by piston up to the point of release: (See Fig. 21.)

(It is convenient to take this point *just before* release.)

Crank-end, as per diagram,	3.73 inches.
In fraction of stroke,	at the end.
Back-end, as per diagram,	3.79 inches.
In fraction of stroke,	at the end.

Volume of cylinder to point of release in cubic feet,

Crank-end, 28.675 + 0.932 =	29.607
Back-end, 29.453 + 0.932 =	30.385

Distance of piston from the end of its stroke at the beginning of compression,

(It is convenient to take this point *just after* compression begins.)

Crank-end, as per diagram,	0.40 inch.
In fraction of stroke, 0.40 ÷ 3.73 =	0.1072
Back-end, as per diagram,	0.4 inch.
In fraction of stroke, 0.40 ÷ 3.79 =	0.1055

Volume of cushion steam from the beginning of compression in cubic feet,

Crank-end, $(0.0325 + 0.1072) \times 28.675 =$ 4.0059
Back-end, $(0.0316 + 0.1055) \times 29.453 =$. 4.0380

Steam pressure at release per square inch above zero,

Crank-end, ... 31.2 pounds.
Back-end, ... 31.2 pounds.

Weight of one cubic foot of steam of the pressure of release,

Crank-end, $p = 31.2$ pounds, . . 0.076,362 lb.
Back-end, $p = 31.2$ pounds, . . 0.076,362 lb.

Steam pressure at the beginning of compression in pounds per square inch above zero,

Crank-end, ... : 22.3 lbs.
Back-end, ... 28.3 lbs.

Weight of one cubic foot of steam of the pressure at the beginning of compression,

Crank-end, $p = 22.3$ pounds, . . 0.055658 lb.
Back-end, $p = 28.3$ pounds, . . 0.069683 lb.

Weight of steam used per stroke,

Crank-end,
$\{(29.607 \times 0.076362)\} - \{(4.006 \times 0.055658)\} =$ 2.0379 lbs.
Back-end,
$\{(30.385 \times 0.076362)\} - \{(4.038 \times 0.069683)\} =$ 2.0389 lbs.

Weight of steam used per hour,

$60 \times 58 \times \{2.0379 + 2.0389\} =$. 14,187.264 lbs.

Weight of steam per horse power per hour,

$14,187.264 \div 650.976 =$ . . . 21.793 lbs.

CASE II. WHEN THE CLEARANCE SPACE IS NOT KNOWN.

The steam exhausted per stroke is the measure of that used per stroke, and may be approximately found by subtracting from the volume of steam at the end of the expansion the volume of steam of the same pressure retained in the cylinder after the exhaust valve is closed.

Continue the expansion line to the point *a*, Figs. 22 and 23, and through this point draw the line *ab* parallel to the zero line, then *ab* represents the volume of steam of the terminal pressure at *a* used per stroke. Of course this takes no account of the water due to condensation contained in the cylinder, nor does any method which depends upon indicator diagrams only.

In the diagrams of the engine under consideration the point *b* falls within the length of the diagram as in Fig. 22.

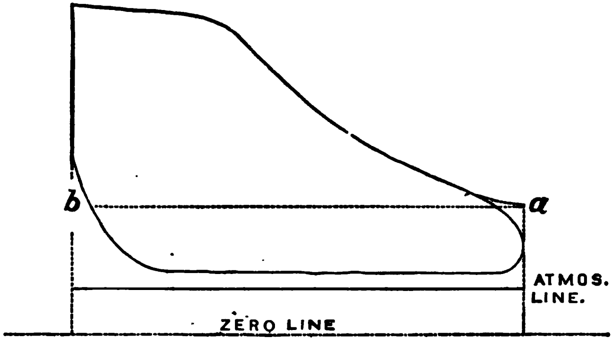


FIG. 22.

Length of diagrams in inches,	
Crank-end,	3·73
Back-end,	3·79
Length of the line *ab* in inches,	
Crank-end,	3·44
Back-end,	3·54

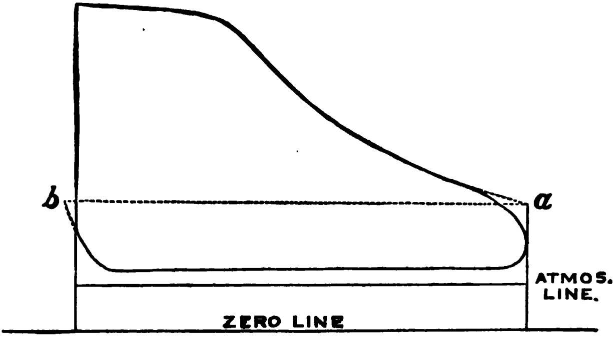


FIG. 23.

Ratio of *ab* to length of diagram, or proportion of piston's stroke,

Crank-end, 3·44 ÷ 3·73 = . . . 0.9222

Back-end, 3·54 ÷ 3·79 = . . . 0.9340

Pressure in pounds per square inch above zero of the steam at the end of expansion, or at the point a,	
Crank-end,	31.2
Back-end,	31.2
Weight of one cubic foot of steam of the above pressures,	
Crank-end, $p = 31.2$ pounds,	0.076,362 lb.
Back-end, $p = 31.2$ pounds,	0.076,362 lb.
Net area of piston in square inches,	
Crank-end,	688.195
Back-end,	706.86
Piston displacement per stroke, in cubic feet,	
Crank-end, $(688.195 \times 72) \div 1728 =$	28.675
Back-end, $(706.86 \times 72) \div 1728 =$	29.453
Volume of steam per stroke of the pressure a, corresponding to the length ab, in cubic feet,	
Crank-end, 0.9222×28.675	26.444
Back-end, 0.9340×29.453	27.509
Weight of steam per stroke of the pressure a,	
Crank-end, 26.444×0.076362	2.0193 lbs.
Back-end, 27.509×0.076362	2.1006 lbs.
Weight of the steam of the pressure a per hour (58 revolutions per minute),	
$60 \times 58 \times \{ 2.0193 + 2.1006 \} =$	14,337.252 lbs.
Weight of steam per horse power per hour,	
$14{,}337.252 \div 650.976$	22.024 lbs.
As obtained before by the more accurate method,	21.793 lbs.
Difference,	0.231 lb.

STEAM CONSUMPTION AS DETERMINED FROM THE INDICATOR DIAGRAMS OF LOW PRESSURE STEAM CYLINDER.

CASE I. WHERE THE CLEARANCE SPACE IS KNOWN.

Net area of piston, in square inches,	
Crank-end,	2441.362
Back-end,	2463.01
Stroke, 72 inches,	6 feet.
Piston displacement per stroke, in cubic feet,	
Crank-end, $\{ 2441.362 \times 72 \} \div 1728$	101.7234
Back-end, $\{ 2463.01 \times 72 \} \div 1728$	102.6254
Clearance space,	
Crank-end, 5884 cubic inches or,	3.405 cub. feet.

In fraction of piston displacement,	
$5884 \div \{2441.362 \times 72\}$	0.0334
Back-end, 5884 cubic inches or,	3.405 cub. feet.
In fraction of piston displacement,	
$5884 \div \{2463.01 \times 72\}$	0.0332
Volume of cylinder in cubic feet,	
Crank-end, $101.7234 + 3.405 =$	105.1284
Back-end, $102.6254 + 3.405 =$	106.0304
Length of diagrams in inches,	
Crank-end,	3.84
Back-end,	3.86
Distance travelled by piston when the steam was released,	
Crank-end, as per diagram, in inches,	3.70
In fraction of stroke, $3.70 \div 3.84 =$	0.9635
Back-end, as per diagram, in inches,	3.75
In fraction of stroke, $3.75 \div 3.86 =$	0.9715
Volume of cylinder to point of release, in cubic feet,	
Crank-end, $(.0334 + 0.9635) \times 101.7234 =$	101.408
In fraction of stroke, $0.0334 + 0.9635$	0.9969
Back-end, $(0.0332 + 0.9715) \times 102.6254$	103.1077
In fraction of stroke, $0.0332 + 0.9715$	1.0047
Distance of piston from end of stroke when compression begins,	
Crank end, as per diagram,	0.45 inch.
In fraction of stroke, $0.45 \div 3.84 =$	0.1172
Back-end, as per diagram,	0.60 inch.
In fraction of stroke, $0.60 \div 3.86 =$	0.1554
Volume of cushion steam from beginning of compression in cubic feet,	
Crank-end, $(0.0334 + 0.1172) \times 101.7234 =$	15.319
In fraction of stroke, $0.0334 + 0.1172$	0.1506
Back-end, $(0.0332 + 0.1554) \times 102.6254 =$	19.355
In fraction of stroke, $0.0332 + 0.1554$	0.1886
Steam pressure at release in pounds per square inch above zero,	
Crank-end,	8.0
Back-end,	8.0
Weight of one cubic foot of steam of the pressure of release,	
Crank-end, $p = 8.0$ pounds.	0.02125 lb.
Back-end, $p = 8.0$ pounds,	0.02125 lb.
Steam pressure at the beginning of compression, in pounds per square inches above zero,	
Crank-end,	2.8
Back-end,	2.4

Weight of one cubic foot of steam at the pressure of beginning of compression,

Crank-end, $p = 2.8$ pounds, . . 0.007904 lb.
Back-end, $p = 2.4$ pounds, . . 0.006832 lb.

Weight of steam per stroke, in pounds,

Crank end,
$(101.408 \times 0.02125) - (15.319 \times 0.007904) =$ 2.0338
Back end,
$(103.1077 \times 0.02125) - (19.355 \times 0.006832) =$ 2.0588

Weight of steam per hour (58 revs. per min.)
$60 \times 58 \times (2.0338 + 2.0588) =$. 14242.248 pounds.

Weight of steam per horse power of whole engine per hour $= 14242.248 \div 1338.617 =$ 10.639 pounds.

Weight of steam per horse power of low pressure cylinder per hour $= 14242.248 \div 687.641 =$ 20.711 lbs.

This may be expressed in algebraic language as follows:

Weight of steam per revolution.

$$\overset{\text{Crank end}}{\left\{\frac{A_c(S_r W_r - S_c W_c)}{144}\right\} S} + \overset{\text{Back end.}}{\left\{\frac{A_b(s_r w_r - s_c w_c)}{144}\right\} S}$$

Weight of steam per hour

$$60 \times R \times \left\{A_c(S_r W_r - S_c W_c) + A_b(s_r w_r - s_c w_c)\right\} \frac{S}{144}$$

which divided by the horse power gives the weight of steam per horse power per hour.

The various letters used above indicate:—

	Crank end	Back end.
Net area of piston in square inches,	A_c	A_b
Fraction $(v_r \div v)$, See Fig. 24.	S_r	s_r
Fraction $v_c \div v$, See Fig. 24.	S_c	s_c
Weight of one cubic foot of steam of release pressure,	W_r	w_r
Weight of one cubic foot of steam of pressure at beginning of compression,	W_c	w_c

$v_l =$ clearance space,
$S =$ length of stroke in feet,
$R =$ number of revolutions per minute.

STEAM PER HORSE POWER PER HOUR CALCULATED FROM THE DIAGRAMS WITHOUT KNOWING THE HORSE POWER, NUMBER OF REVOLUTIONS, OR THE LENGTH OF THE STROKE.

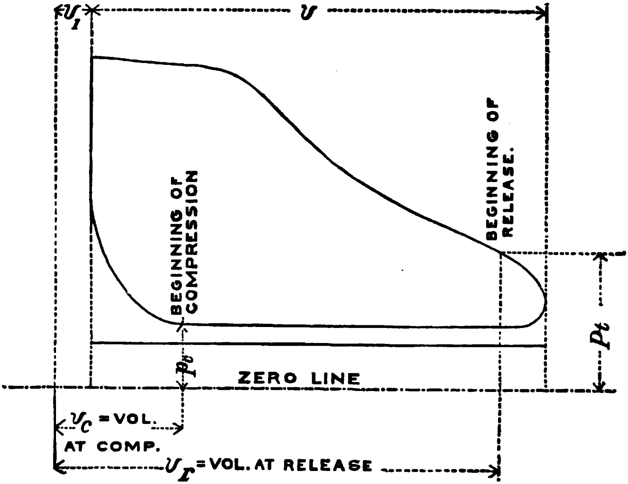


FIG. 24.

The horse power is $$\frac{(\overset{\text{Crank.}}{A_c p_c} + \overset{\text{Back.}}{A_b p_b}) \times R \times S}{33,000}$$

in which p_c and p_b indicate the mean effective pressure of the crank and back ends respectively.

Then the steam per hour divided by this, or

$$60 \times R \times \left\{ A_c (S_r W_r - S_c W_c) + A_b (s_r w_r - s_c w_c) \right\} \frac{S}{144}$$

divided by $$\frac{R \times S \times (A_c p_c + A_b p_b)}{33,000}$$

or

$$\frac{13,750 \left\{ A_c (S_r W_r - S_c W_c) + A_b (s_r w_r - s_c w_c) \right\}}{A_c p_c + A_b p_b}$$

Example. Low pressure steam cylinder.

	Crank end.	Back end.
Net area of piston in square inches,	2441.362	2463.01

Fraction $v_r \div v$, . . .	0.9969	1.0047
Fraction $v_c \div v$, . . .	0.1506	0.1886
Weight of cubic foot of steam at release, ...	0.02125	0.02125
Weight of cubic foot of steam at compression, . . .	0.007904	0.006832
Mean effective pressure, . .	13.12 lbs.	13.47 lbs.

From the above the weight of steam accounted for per horse power per hour of the low pressure cylinder by the diagrams of that cylinder will be :—

$$13{,}750\left\{2{,}441.362\,\{(0.9969 \times 0.02125) - (0.1506 \times 0.007904)\} + 2{,}463.01\,\{(1.0047 \times 0.02125) - (0.1886 \times 0.006832)\}\right\}$$

divided by $(2441.362 \times 13.12) + (2463.01 \times 13.47)$

$$= 20.711 \text{ pounds.}$$

the same as before.

Or, if we wish the steam per horse power of the whole engine we must use for the divisor (see horse power of high pressure cylinder)

Low Pressure Cylinder.

$$[(2{,}441.362 \times 13.12) + (2{,}463.01 \times 13.47)] +$$

High Pressure Cylinder.

$$[(688.195 \times 45.04) + (706.86 \times 43.48)]$$

or $65{,}207.41414 + 61{,}730.5756 = 126{,}937.98974$

Dividing by this we have, as before, . . 10.639 pounds.

CASE II. WHEN THE CLEARANCE SPACE IS UNKNOWN.

In this, as in the case of the high pressure steam cylinder, the length *ab* is less than the length of the diagram.

Length of diagrams in inches,	
Crank end, ...	3.84
Back end, ...	3.86
Length of line *ab* (see Fig. 25) in inches,	
Crank end, ...	3.80
Back end, ...	3.80
Ratio of *ab* to length of diagram, or proportion of piston's stroke,	
Crank end, 3.80 ÷ 3.84, . . .	0.9895
Back end, 3.80 ÷ 3.86, . . .	0.9844
Pressure, in pounds per sq. in. above zero, of the steam at the end of expansion, or, at the point *a*,	

Crank end, ...	7.0
Back end, ...	7.8
Weight of one cubic foot of steam of the above pressures,	
Crank end, $p = 7.0$, ...	0.01874 pound.
Back end, $p = 7.8$, . . .	0.020748 pound.

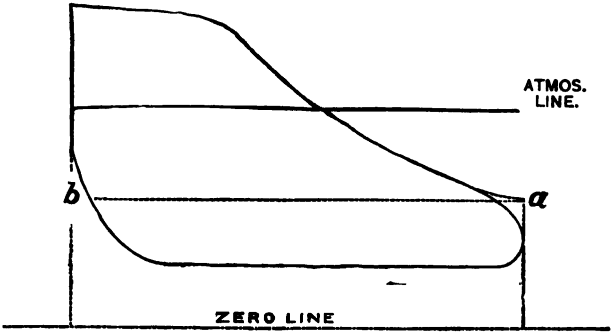


Fig. 25.

Net area of piston in sq. in.,	
Crank end, ...	2,441.362
Back end, ...	2,463.01
Piston displacement per stroke, in cubic feet,	
Crank end, (72 × 2,441.362) ÷ 1,728 =	101.7234
Back end, (72 × 2,463.01) ÷ 1,728 =	102.6254
Volume of steam per stroke of the pressure *a*, corresponding to the length *ab*, in cubic feet,	
Crank end, 0.9895 × 101.7234 = . .	100.655
Back end, 0.9844 × 102.6254 = . .	101.0244
Weight of steam per stroke of the pressure *a*,	
Crank end, 100.655 × 0.01874 = . .	1.8862
Back end, 101.024 × 0.020748 = . .	2.0960

Weight of steam of the pressure *a* per hour (58 revolutions per minute),
$60 \times 58 \times (1.8862 + 2.0960) =$ . . . 13,858.056 lbs.

Horse power of both high and low pressure cylinders, ... 1,338.617

Weight of steam per horse power per hour,
$13{,}858.056 \div 1{,}338.617 =$ . . . 10.352 lbs.

Weight as determined by CASE 1, . . 10.639 lbs.

Difference, 10.639 — 10.352, . . . 0.287 lb.

Weight of steam per horse power of low pressure cylinder per hour,
$13{,}858.056 \div 687.641 =$ . . . 20.15 pounds.

The preceding calculation may also be made without knowing the clearance space, horse power, number of revolutions or the length of stroke, as follows:—

Volume of steam per stroke in cubic feet,

Crank end, $\dfrac{72 \times 2{,}441.362}{1.728} \times \dfrac{3.80}{3.84}$

Back end, $\dfrac{72 \times 2{,}463.01}{1{,}728} \times \dfrac{3.80}{3.86}$

Weight of steam per stroke,

Crank end $\dfrac{72 \times 2{,}441.362}{1{,}728} \times \dfrac{3.80}{3.84} \times 0.01874$ pounds.

Back end, $\dfrac{72 \times 2{,}463.01}{1{,}728} \times \dfrac{3.80}{3.86} \times 0.020748$ pounds.

Weight of steam per hour,

$$\frac{60 \times 58}{1{,}728} \left\{ \left(72 \times 2{,}441.362 \times \frac{3.80}{3.84} \times 0.01874\right) + \left(72 \times 2{,}463.01 \times \frac{3.80}{3.86} \times 0.020748\right) \right\}$$

Mean effective pressures in low pressure cylinder,
Crank end, ... 13.12 lbs.
Back end, ... 13.47 lbs.

Horse power of the low pressure cylinder,

$$\frac{\left\{ (2{,}441.362 \times 13.12) + (2{,}463.01 \times 13.47) \right\} \times 58 \times 6}{33{,}000}$$

The steam per horse power per hour will then be:

$$\frac{60 \times 58 \times 72}{1728} \left\{ \left(2{,}441.362 \times \frac{3.80}{3.84} \times 0.01874\right) + \left(2{,}463.01 \times \frac{3.80}{3.86} \times 0.020748\right) \right\}$$

divided by

$$\frac{\{(2{,}441.362 \times 13.12)+(2{,}463.01 \times 13.47)\} \times 58 \times 6}{33{,}000}$$

or,

$$13{,}750\left\{ (2{,}441.362 \times \frac{3.80}{3.84} \times 0.01874) + (2{,}463.01 \times \frac{3.80}{3.86} \times 0.020748) \right\}$$

divided by

$$(2{,}441.362 \times 13.12) + (2{,}463.01 \times 13.47)$$

or,

$$13{,}750\{ (2{,}441.362 \times 0.9895 \times 0.01874) + (2{,}463.01 \times 0.9844 \times 0.020748) \}$$

divided by

$$(2{,}441.362 \times 13.12)+(2{,}463.01 \times 13.47)$$

or 20.15 pounds, the same as before.

This may be expressed in algebraic language as follows: Weight of steam per horse power per hour,

$$13{,}750\frac{\{ A s w + A_{,} s_{,} w_{,} \}}{A p + A_{,} p_{,}} \quad \text{in which}$$

A and $A_{,}$ = net areas of piston at opposite ends.

s and $s_{,}$ = ratio of line *ab* to length of diagram.

w and $w_{,}$ = weight of one cubic foot of steam of the *absolute* terminal pressure at *a*.

p and $p_{,}$ = M. E. P. obtained from the diagram.

WEIGHT OF STEAM IN THE LOW PRESSURE CYLINDER AT CUT-OFF AS DETERMINED FROM THE DIAGRAMS.

Pressure at cut-off in pounds per square inch above zero,	
Crank end, ...	18.6
Back end, ...	18.8
Weight of one cubic foot of steam corresponding to the pressure at cut-off,	
Crank end, ...	0.046904 pound.
Back end, ...	0.047382 pound.
Distance of piston from beginning of stroke at cut-off,	
Crank end, as per diagram, . .	1.5 inches,
In fraction of stroke 1.5 ÷ 3.84 , ,	0.3906

Back end, as per diagram,	1.6 inches.
In fraction of stroke $1.6 \div 3.86$	0.4145
Displacement of piston per stroke in cubic feet,	
Crank end, $(2{,}441.362 \times 72) \div 1{,}728 =$	101.7234
Back end, $(2{,}463.01 \times 72) \div 1{,}728 =$	102.6254
Clearance space,	
Crank end = 5,884 cubic inches or	3.405 cub. ft.
In fraction of piston displacement,	
$5{,}884 \div (2{,}441{,}362 \times 72)$	0.0334
Back end, 5,884 cubic inches =	3.0405 cub. ft.
In fraction of stroke	
$5{,}884 \div (2{,}463.01 \times 72)$	0.0332
Volume of steam in cylinder at cut-off in cubic feet,	
Crank end, $(0.3906 + 0.0334) \times 101.7234 =$	43.1307
Back end, $(0.4145 + 0.0332) \times 102.6254 =$	45.9454
Weight of steam in cylinder at cut-off,	
Crank end, $43.1307 \times 0.046904 =$	2.0230 pounds.
Back end, $45.9454 \times 0.047382 =$	2.1770 pounds.

WEIGHT OF STEAM CONSUMED PER HORSE POWER PER HOUR AS DETERMINED FROM THE WEIGHT OF STEAM IN THE CYLINDER AT CUT-OFF AS ACCOUNTED FOR BY THE DIAGRAMS FROM THE LOW PRESSURE CYLINDER.

Weight of cushion steam per stroke from the beginning of compression,	
Crank end, $15.319 \times 0.007904 =$	0.1211
Back end, $19.355 \times 0.006832 =$	0.1322
Weight of steam used per stroke,	
Crank end, $2.023 - 0.1211 =$	1.9019 lbs.
Back end, $2.177 - 0.1322 =$	2,0448 lbs.
Weight of steam used per hour,	
$60 \times 58 \times \{ 1.9019 + 2.0448 \} =$	13,734.516 lbs.
Weight of steam used per horse power per hour,	
$13{,}734.516 \div 1{,}338.617 =$	10.26 lbs.

USE OF LOGARITHMS.

Every engineer should know the use of logarithms, as they save much labor in calculations and are very easily learned. An example showing the saving of work in calculating the horse power of the engine will illustrate this.

For the horse power of the high pressure we have (see page 151):

$$\{(688.195 \times 45.04) + (706.86 \times 43.48)\} \times \frac{58 \times 6}{33000}$$

$$\begin{array}{rl} 58 \times 6 = 348 & \log. = 2.5415792 \\ 33000 & \log. = \underline{4.5185139} \\ 348 \div 33000 & \log. = \bar{2}.0230653 \\ 688.195 & \log. = 2.8377115 \\ 45.04 & \log. = \underline{1.6535984} \end{array}$$

H. P. of Crank End

$$(688.195 \times 45.04) \times \frac{58 \times 6}{33000} = 2.5143752 = \quad 326.87$$

$$\begin{array}{rl} 348 \div 33000 & \log. = \bar{2}.0230653 \\ 706.86 & \log. = 2.8493334 \\ 43.48 & \log. = \underline{1.6382895} \end{array}$$

H. P. of Back End

$$(706.86 \times 43.48) \times \frac{58 \times 6}{33000} = 2.5106882 = \quad \underline{324.10}$$

Horse power of high pressure cylinder, as before, = 650.97

For the horse power of the low pressure cylinder we have (see page 152):

$$\{(2,441.362 \times 13.12) + (2,463.01 \times 13.47)\} \times \frac{58 \times 6}{33000}$$

$$\begin{array}{rl} 348 \div 33000 & \log. \ \bar{2}.0230653 \\ 2,441.362 & \log. \ 3.3876321 \\ 13.12 & \log. \ \underline{1.1179338} \end{array}$$

H. P. of Crank End

$$(2,441.362 \times 13.12) \times \frac{58 \times 6}{33000} = 2.5286312 = \quad 337.78$$

$$\begin{array}{rl} 348 \div 33000 & \log. \ \bar{2}.0230653 \\ 2,463.01 & \log. \ 3.3914662 \\ 13.47 & \log. \ \underline{1.1293676} \end{array}$$

H. P. of Back End

$$(2,463.01 \times 13.47) \times \frac{58 \times 6}{33000} = 2.5438991 = \quad \underline{349.86}$$

Horse power of low pressure cylinder as before = 687.64

AREAS OF CIRCLES.

Diam.	Area.	Diam.	Area.	Diam.	Area.	Diam.	Area.	Diam.	Area.
½	.1963	13	132.7	28	615.8	46	1662	76	4537
9/16	.2485	13¼	137.9	28¼	626.8	46½	1698	76½	4596
⅝	.3068	13½	143.1	28½	637.9	47	1735	77	4657
11/16	.3712	13¾	148.5	28¾	649.2	47½	1772	77½	4717
¾	.4418	14	153.9	29	660.5	48	1810	78	4778
13/16	.5185	14¼	159.5	29¼	672.0	48½	1847	78½	4840
⅞	.6013	14½	165.1	29½	683.5	49	1886	79	4902
15/16	.6903	14¾	170.9	29¾	695.1	49½	1924	79½	4963
1	.7854	15	176.7	30	706.9	50	1964	80	5027
1⅛	.9940	15¼	182.6	30¼	718.7	50½	2003	80½	5090
1¼	1.227	15½	188.7	30½	730.6	51	2043	81	5153
1⅜	1.485	15¾	194.8	30¾	742.6	51½	2083	81½	5217
1½	1.767	16	201.1	31	754.8	52	2124	82	5281
1⅝	2.074	16¼	207.4	31¼	767.0	52½	2165	82½	5346
1¾	2.405	16½	213.8	31½	779.3	53	2206	83	5411
1⅞	2.761	16¾	220.4	31¾	791.7	53½	2248	83½	5476
2	3.142	17	227.0	32	804.2	54	2290	84	5542
2¼	3.976	17¼	233.7	32¼	816.9	54½	2333	84½	5608
2½	4.909	17½	240.5	32½	829.6	55	2375	85	5675
2¾	5.940	17¾	247.5	32¾	842.4	55½	2419	85½	5742
3	7.069	18	254.5	33	855.3	56	2463	86	5809
3¼	8.296	18¼	261.6	33¼	868.3	56½	2507	86½	5877
3½	9.621	18½	268.8	33½	881.4	57	2552	87	5945
3¾	11.04	18¾	276.1	33¾	894.6	57½	2597	87½	6013
4	12.57	19	283.5	34	907.9	58	2642	88	6082
4¼	14.19	19¼	291.0	34¼	921.3	58½	2688	88½	6151
4½	15.90	19½	298.6	34½	934.8	59	2734	89	6221
4¾	17.72	19¾	306.4	34¾	948.4	59½	2781	89½	6291
5	19.64	20	314.2	35	962.1	60	2827	90	6362
5¼	21.65	20¼	322.1	35¼	975.9	60½	2875	90½	6432
5½	23.75	20½	330.1	35½	989.8	61	2922	91	6504
5¾	25.97	20¾	338.2	35¾	1004	61½	2971	91½	6576
6	28.27	21	346.4	36	1018	62	3019	92	6648
6¼	30.68	21¼	354.7	36¼	1032	62½	3068	92½	6720
6½	33.18	21½	363.1	36½	1046	63	3117	93	6793
6¾	35.78	21¾	371.5	36¾	1061	63½	3167	93½	6866
7	38.48	22	380.1	37	1075	64	3217	94	6940
7¼	41.28	22¼	388.8	37¼	1090	64½	3268	94½	7014
7½	44.18	22½	397.6	37½	1105	65	3318	95	7088
7¾	47.17	22¾	406.5	37¾	1120	65½	3370	95½	7163
8	50.27	23	415.5	38	1134	66	3421	96	7238
8¼	53.46	23¼	424.6	38¼	1149	66½	3474	96½	7314
8½	56.75	23½	433.7	38½	1164	67	3526	97	7390
8¾	60.1	23¾	443.0	38¾	1179	67½	3578	97½	7466
9	63.62	24	452.4	39	1195	68	3632	98	7543
9¼	67.20	24¼	461.9	39¼	1210	68½	3685	98½	7620
9½	70.88	24½	471.4	39½	1225	69	3739	99	7697
9¾	74.66	24¾	481.1	39¾	1241	69½	3794	99½	7776
10	78.54	25	490.9	40	1257	70	3848	100	7854
10¼	82.52	25¼	500.7	40½	1288	70½	3904	101	8011
10½	86.59	25½	510.7	41	1320	71	3959	102	8171
10¾	90.76	25¾	520.8	41½	1352	71½	4015	103	8332
11	95.03	26	530.9	42	1385	72	4072	104	8495
11¼	99.50	26¼	541.2	42½	1418	72½	4128	105	8659
11½	103.9	26½	551.5	43	1452	73	4185	106	8825
11¾	108.4	26¾	562.0	43½	1487	73½	4243	107	8992
12	113.1	27	572.6	44	1521	74	4301	108	9161
12¼	117.9	27¼	583.2	44½	1555	74½	4359	109	9331
12½	122.7	27½	594.0	45	1590	75	4418	110	9503
12¾	127.7	27¾	604.8	45½	1626	75½	4477	111	9677

LOGARITHMS.

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts.								
											1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

LOGARITHMS.

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts.								
											1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	5159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	7370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

WATER BETWEEN 32° AND 212° F.

Temperature Fahr.	Heat Units per lb.	Weight, lbs. per cubic foot.	Temperature Fahr.	Heat Units per lb.	Weight, lbs. per cubic foot.	Temperature Fahr.	Heat Units per lb.	Weight, lbs. per cubic foot.
32°	0.00	62.42	123°	91.09	61.68	168°	136.45	60.81
35	3.02	62.42	124	92.10	61.67	169	137.46	60.79
40	8.06	62.42	125	93.10	61.65	170	138.46	60.77
45	13.08	62.42	126	94.11	61.63	171	139.47	60.75
50	18.10	62.41	127	95.12	61.61	172	140.48	60.73
52	20.11	62.40	128	96.13	61.60	173	141.49	60.70
54	22.11	62.40	129	97.14	61.58	174	142.50	60.68
56	24.11	62.39	130	98.14	61.56	175	143.50	60.66
58	26.12	62.38	131	99.15	61.54	176	144.51	60.64
60	28.12	62.37	132	100.16	61.52	177	145.52	60.62
62	30.12	62.36	133	101.17	61.51	178	146.53	60.59
64	32.12	62.35	134	102.18	61.49	179	147.54	60.57
66	34.12	62.34	135	103.18	61.47	180	148.54	60.55
68	36.12	62.33	136	104.19	61.45	181	149.55	60.53
70	38.11	62.31	137	105.20	61.43	182	150.56	60.50
72	40.11	62.30	138	106.21	61.41	183	151.57	60.48
74	42.11	62.28	139	107.22	61.39	184	152.58	60.46
76	44.11	62.27	140	108.22	61.37	185	153.58	60.44
78	46.10	62.25	141	109.23	61.36	186	154.59	60.41
80	48.09	62.23	142	110.24	61.34	187	155.60	60.39
82	50.08	62.21	143	111.25	61.32	188	156.61	60.37
84	52.07	62.19	144	112.26	61.30	189	157.62	60.34
86	54.06	62.17	145	113.26	61.28	190	158.62	60.32
88	56.05	62.15	146	114.27	61.26	191	159.63	60.29
90	58.04	62.13	147	115.28	61.24	192	160.63	60.27
92	60.03	62.11	148	116.29	61.22	193	161.64	60.25
94	62.02	62.09	149	117.30	61.20	194	162.65	60.22
96	64.01	62.07	150	118.30	61.18	195	163.66	60.20
98	66.01	62.05	151	119.31	61.16	196	164.66	60.17
100	68.01	62.02	152	120.32	61.14	197	165.67	60.15
102	70.00	62.00	153	121.33	61.12	198	166.68	60.12
104	72.00	61.97	154	122.34	61.10	199	167.69	60.10
106	74.00	61.95	155	123.34	61.08	200	168.70	60.07
108	76.00	61.92	156	124.35	61.06	201	169.70	60.05
110	78.00	61.89	157	125.36	61.04	202	170.71	60.02
112	80.00	61.86	158	126.37	61.02	203	171.72	60.00
113	81.01	61.84	159	127.38	61.00	204	172.73	59.97
114	82.02	61.83	160	128.38	60.98	205	173.74	59.95
115	83.02	61.82	161	129.39	60.96	206	174.74	59.92
116	84.03	61.80	162	130.40	60.94	207	175.75	59.89
117	85.04	61.78	163	131.41	60.92	208	176.76	59.87
118	86.05	61.77	164	132.42	60.90	209	177.77	59.84
119	87.06	61.75	165	133.42	60.87	210	178.78	59.82
120	88.06	61.74	166	134.43	60.85	211	179.78	59.79
121	89.07	61.72	167	135.44	60.83	212	180.79	59.76
122	90.08	61.70						

WEIGHT OF A CUBIC FOOT OF WATER AT HIGH TEMPERATURES.

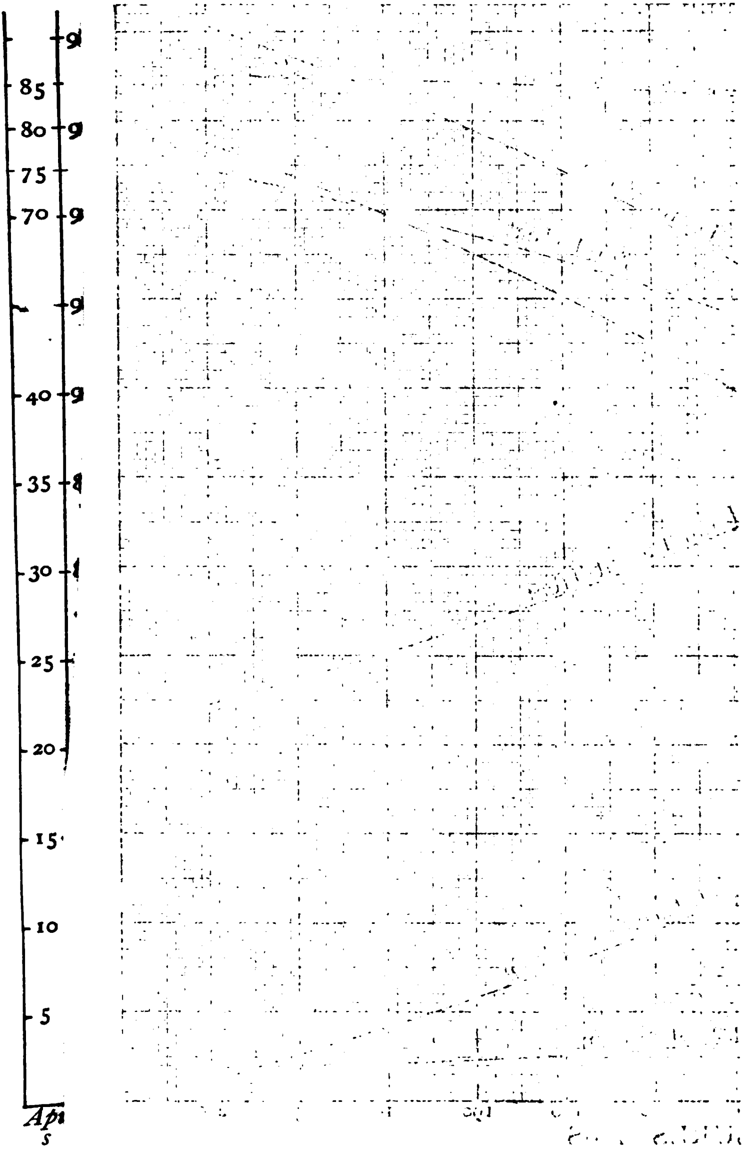
Temperature Fahr.	Weight of 1 cubic foot.	Difference per 1 degree.	Temperature Fahr.	Weight of 1 cubic foot.	Difference per 1 degree.	Temperature Fahr.	Weight of 1 cubic foot.	Difference per 1 degree.
170°	60.800	——	250°	58.812	.0284	330°	56.236	.0348
180	60.587	.0213	260	58.517	.0295	340	55.883	.0353
190	60.366	.0221	270	58.214	.0303	350	55.523	.0360
200	60.136	.0230	280	57.903	.0311	360	55.158	.0365
210	59.894	.0242	290	57.585	.0318	370	54.787	.0371
220	59.641	.0253	300	57.259	.0326	380	54.411	.0376
230	59.372	.0269	310	56.925	.0334	390	54.030	.0381
240	59.096	.0276	320	56.584	.0341	400	53.635	.0395

PHYSICAL PROPERTIES OF SATURATED STEAM BELOW ATMOSPHERIC PRESSURE.

Pressure.		Temperature.	Heat Required to Convert a Pound of Water at 32° F. into Steam.					
Vacuum by Gage. Inches Mercury	Abs'lute Pressure. Pounds per Sq. In.	t Degrees F.	To Raise the Temperature to the Boiling Point. q Heat Units.	To Convert the Water at the Boiling Point into Steam. r Heat Units.	Total. λ Heat Units.	Relative Volume. Water at 39° = 1.	Volume of One Pound. s Cubic Feet.	Weight of One Cubic Foot. Pounds.
1	2	3	4	5	6	7	8	9
29.74	.09	32	0	1091.70	1091.70	211,433	3387	.00030
29.72	.10	35	3	1089.62	1092.62	188,648	3022	.00033
29.67	.12	40	8	1086.14	1094.14	156,437	2506	.00040
29.62	.15	45	13	1082.66	1095.66	130,281	2087	.00048
29.56	.18	50	18	1079.19	1097.19	108,932	1745	.00058
29.49	.21	55	23.01	1075.71	1098.72	91,453	1465	.00068
29.40	.25	60	28 01	1072.23	1100.24	77,032	1234	.00082
29.30	.30	65	33.01	1068.75	1001.76	65,047	1042	.00096
29.19	.36	70	38.02	1065.27	1103.29	53,373	885	.00115
29	.43	75	43.03	1061.79	1104.82	47,037	753 50	.00133
28.90	.50	80	48.04	1058 30	1106.34	40,189	643.80	.00158
28.72	.59	85	53.05	1054.82	1107.87	34,440	551.70	.00181
28.51	.69	90	58.06	1051.36	1109.42	29,627	474 60	.00213
28.27	.81	95	63.07	1047.85	1110.92	25,505	409.30	.00244
28	.94	100	68.08	1044.36	1112.44	22,098	354	.00286
27.89	1	102	70.09	1042.96	1113.05	20,881	334.50	.00299
27.69	1.09	105	73.10	1040.87	1113.97	19,183	307.30	.00325
27.34	1.27	110	78.11	1037.38	1115.49	16,699	267.50	.00374
27	1.44	114.34	82.47	1034.34	1116.81	14,772	236.64	.00423
26.95	1.46	115	83.13	1033.89	1117.02	14,576	233.50	.00428
26.50	1.68	120	88.15	1030.39	1118.54	12,760	204.40	.00489
26	1.93	125	93.17	1026.80	1120.07	11,200	179.40	.00557
25.85	2	126.27	94.44	1026.01	1120.45	10,830	173.49	.00576
25.42	2.21	130	98.19	1023.39	1121.58	9,851	157.80	.00634
25	2.42	133	101.21	1021.30	1122.51	9,139	146.40	00683
24.79	2.53	135	103.22	1019.90	1123.12	8,689	139.20	.00718
24.06	2.88	140	108.25	1016.40	1124.65	7,691	123.20	.00812
24	2.90	141.29	109.54	1015.49	1125.03	7,638	122.36	.00817
23.81	3	141.62	109.88	1015.25	1125.13	7,388	118.35	.00845
23.26	3.27	145	113.28	1012.89	1126.17	6,817	109.20	.00916
23	3.40	146.53	114.81	1011.82	1126.63	6,565	105.17	.00951
22.37	3.71	150	118.31	1009.39	1127.70	6,057	97.03	.01031
22	3.90	152.05	120.37	1007.95	1128.32	5,770	92.43	.01082
21.78	4	153.07	121.40	1007.23	1128.63	5,634	90.26	.01108
21.39	4.19	155	123.34	1005.88	1129.22	5,395	86.42	.01157
21	4.37	156.74	125.09	1004.45	1129.54	5,183	83.03	.01204
20.29	4.73	160	128.37	1002.37	1130.74	4,813	77.10	.01296
20	4.86	161.16	129.54	1001.55	1131.09	4,692	75.16	.01330
19.74	5	162.33	130.72	1000.73	1131.45	4,568	73.17	.01368

PHYSICAL PROPERTIES OF SATURATED STEAM BELOW ATMOSPHERIC PRESSURE.

Pressure.		Temperature.	Heat Required to Convert a Pound of Water at 32° F. into Steam.			Relative Volume. Water at 39° = 1.	Volume of One Pound. *s* Cubic Feet.	Weight of One Cubic Foot. Pounds.
Vacuum by Gage. Inches Mercury	Abs'lute Pressure. Pounds per Sq. In.	*t* Degrees F.	To Raise the Temperature to the Boiling Point. *q* Heat Units.	To Convert the Water at the Boiling Point into Steam. *r* Heat Units.	Total. λ Heat Units.			
1	2	3	4	5	6	7	8	9
19.08	5.32	165	133.41	998.85	1132.26	4,307	69	.01449
19	5.36	165.32	133.73	998.63	1132.36	4,278	68.54	.01459
18	5.86	169	137.45	996.04	1133.49	3,945	63.20	.01582
17.74	5.98	170	138.45	995.34	1133.79	3,864	61.90	.01616
17.71	6	170.12	138.58	995.25	1133.83	3,848	61.64	.01622
17	6.35	172.60	141.07	993.51	1134.58	3,649	58 45	.01711
16.27	6.70	175	143.50	991.82	1135.32	3,471	55.60	.01799
16	6.84	175.90	144.40	991.18	1135.58	3,402	54.50	.01835
15.67	7	176.91	145.43	990.47	1135.90	3,329	53.33	.01875
15	7.33	178.98	147.51	989.02	1136.53	3,188	51.07	.01958
14.65	7.50	180	148.54	988.30	1136 84	3,121	50	.02000
14	7.82	181.88	150.43	986.98	1137.41	3,000	48.06	02081
13.63	8	182.91	151.48	986.25	1137.73	2,934	47.04	.02126
13	8.31	184.67	153.25	985.01	1138.26	2,833	45 39	.02203
12.87	8.38	185	153.59	984.77	1138.36	2,815	45.10	.02217
12	8.80	187.27	155.89	983.36	1139.25	2,685	43.01	.02325
11.60	9	188.32	156.94	982.43	1139.37	2,629	42.11	.02375
11	9.29	189.82	158.46	981.38	1139.84	2,551	40 87	.02447
10.92	9.33	190	158 64	981.25	1139.89	2,541	40.70	.02457
10	9.78	192.21	160.89	979.67	1140.56	2,431	38 94	.02568
9.56	10	193.24	161.92	978 96	1140.88	2,381	38.14	.02622
9	10.28	194.53	163.22	978.05	1141.27	2,316	37.16	.02691
8.79	10.38	195	163.70	977.72	1141.42	2,303	36 90	.02710
8	10.77	196.74	165.46	976.48	1141.94	2,220	35.56	.02812
7.53	11	197.77	166 50	975.76	1142.26	2,176	34.86	.02869
7	11.26	198.90	167 64	974.96	1142.60	2,129	34.10	.02933
6.46	11.52	200	168.75	974.19	1142.94	2,085	33.40	.02994
6	11.70	200.75	169.51	973.65	1143.16	2,054	32 90	.03040
5.49	12	201.96	170.74	972.80	1143.54	2,005	32.12	.03313
5	12.24	202.92	171.71	972.12	1143.83	1,968	31.52	.03173
4	12.73	204.77	173.58	970.82	1144.40	1,896	30 38	.03292
3.93	12.77	205	173 81	970.65	1144.46	1,891	30.30	.03300
3.45	13	205.89	174.71	970.03	1144.74	1,862	29.82	.03355
3	13.22	206.72	175.56	969.43	1144.99	1,822	29.18	.03427
2	13.71	208.52	177.38	968.16	1145 54	1,755	28.11	.03557
1.42	14	209.56	178.43	967.43	1145.86	1,735	27.79	.03600
1.17	14.12	210	178.87	967.12	1145.99	1,721	27.57	.03628
1	14.21	210.30	179.17	966.91	1146.08	1,700	27.23	.03672
....	14.70	212	180.90	965.70	1146.60	1,661	26.60	.03759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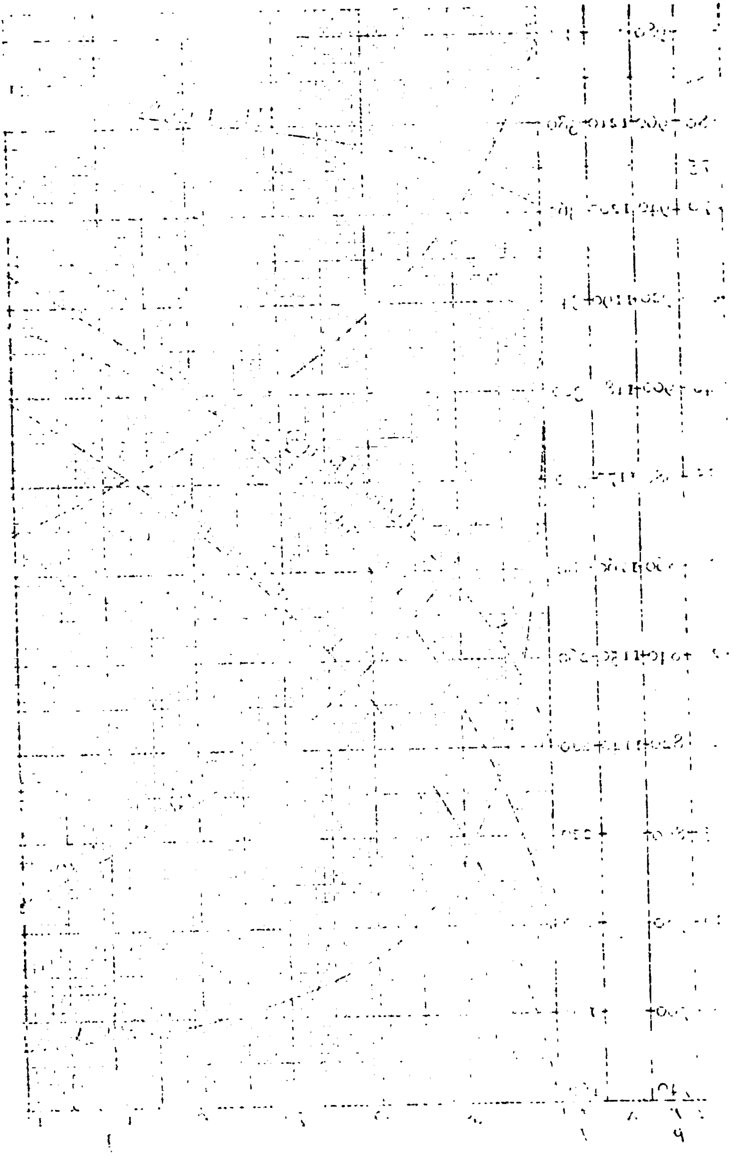

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